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MACHINE DESIGN

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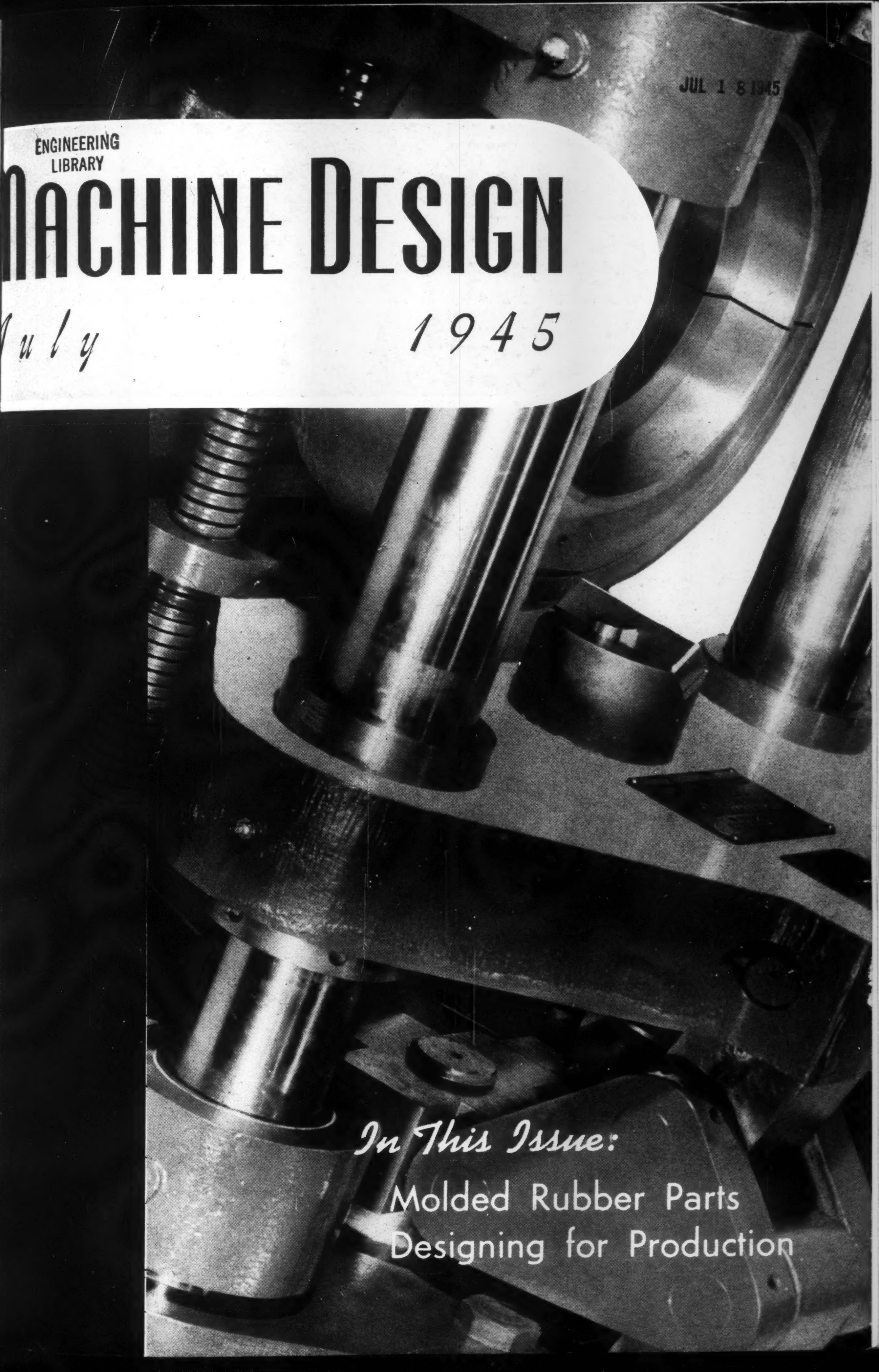

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Molded Rubber Parts
Designing for Production

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# MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

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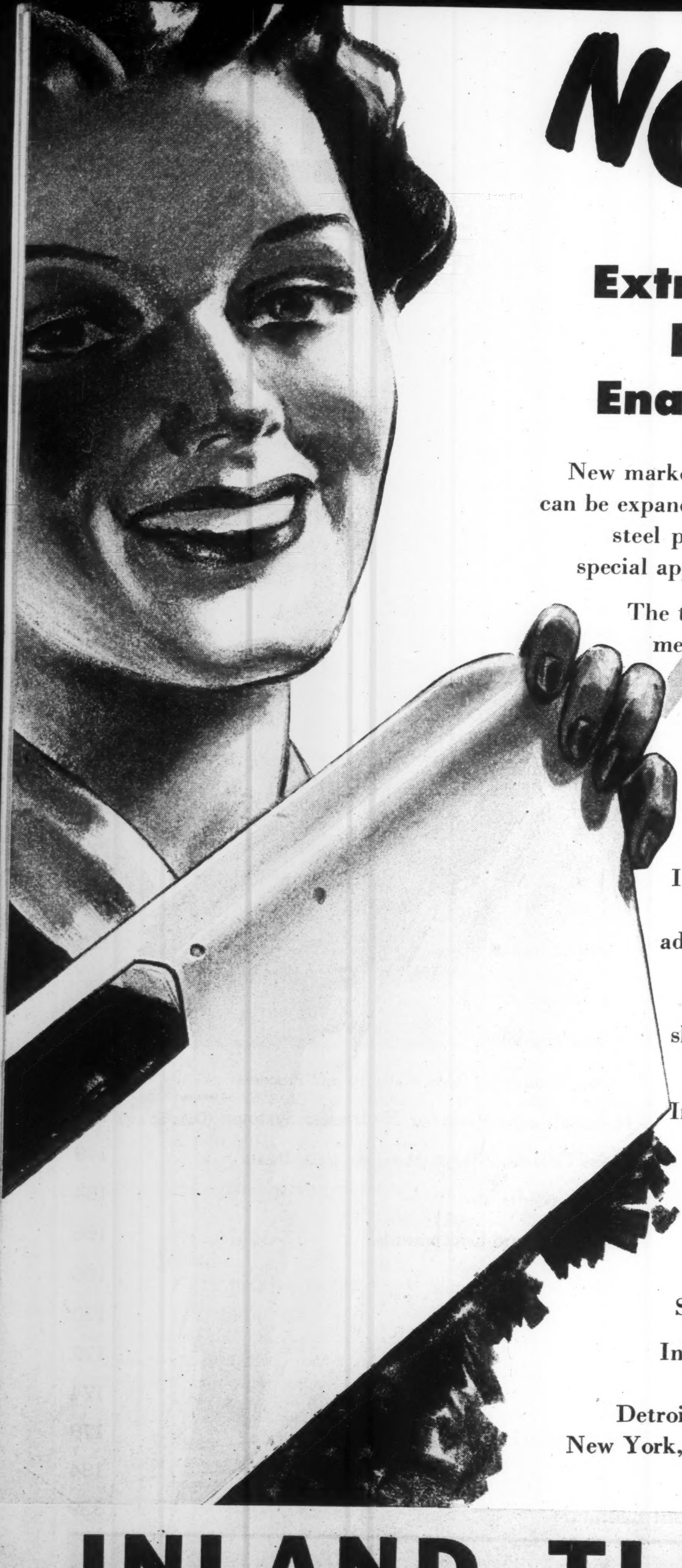

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# INLAND TI-NAMEL

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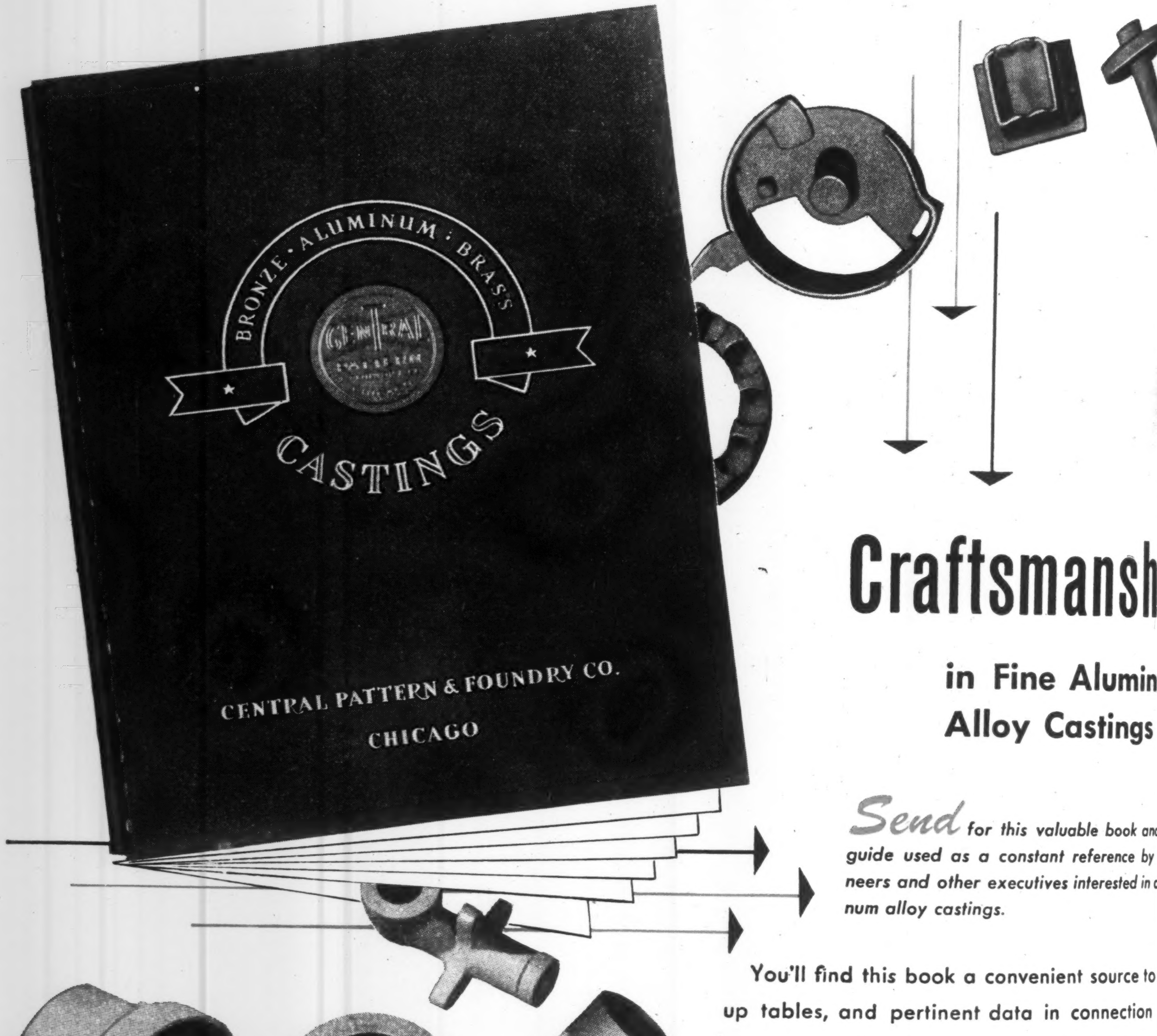

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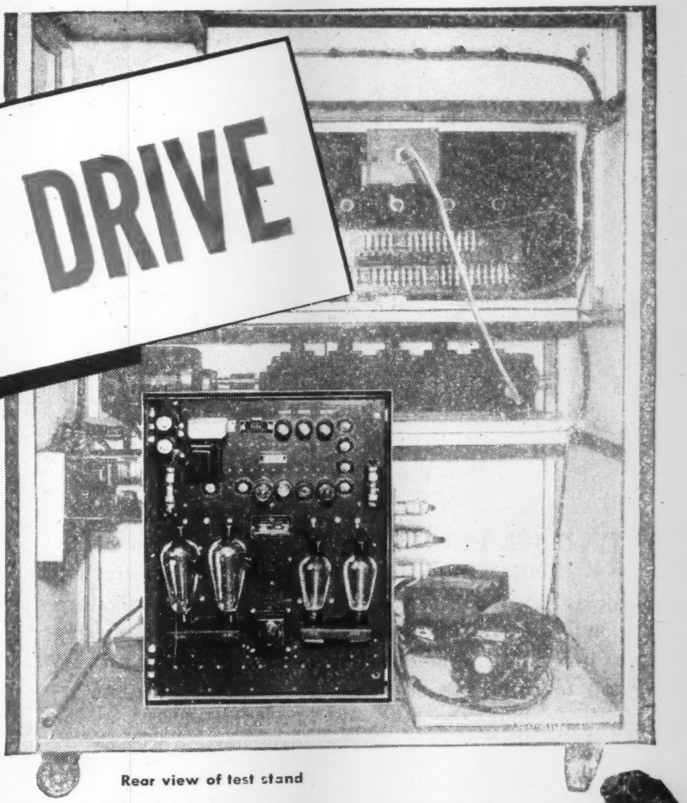

Rear view of test stand

## GIVES A WIDE RANGE OF SPEEDS FOR TACHOMETER TESTS AT LOCKHEED

**At Lockheed, the Thy-mo-trol electronic panel (shown above) is installed in the back of the test stand, and is readily accessible for inspection. The control dials are mounted on the front of the stand, and are easily adjusted by the operator.**

Wide speed range, increased accuracy, and simpli- ed operation were the results of installing G-E hy-mo-trol drive on this tachometer generator test tand of the Lockheed Aircraft Corporation.

Five tachometer generators, all driven by the $\frac{1}{4}$-hp hy-mo-trol drive, can be tested simultaneously. A imple turn of the Thy-mo-trol control dial gives ach of the desired speeds during testing, and the bility to hold speed precisely adds to the accuracy f the test. Test readings are made every 200 rpm, etween 200 to 4000 rpm, by comparing the tachom- ter indicators with a master stroboscope disk.

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# Topics

DIESEL ENGINE - SUPERCHARGER - TURBINE combinations for heavy-duty motor vehicles have been proposed by the Elliott Co. Future possibilities include utilization of the diesel merely to function as a combustion chamber for the turbine and to drive the supercharger with the turbine itself becoming the source of power.

DROOP-SNOOT planes fooled the Germans for a year and tricked their Luftwaffe into offering no fighter opposition to scores of successful bombing missions. Because the Germans avoided combat with fighters—conserving their dwindling strength against bombers—the Americans modified the P38 fighter by adding a droop-snoot nose to carry a bombardier with Norden bombsight and 4000 pounds in bombs.

DIESEL ENGINES cannot rest on present laurels of fuel economy, the advantage being lost unless related costs of maintenance can be held at the level of competitive power sources. For mobile applications the diesel has been less satisfactory than for stationary engines because of engineering problems not yet solved.

NEW SERVICE to industry and inventors has been instituted by the United States Patent Office in a Register of Patents Available for Licensing. The purpose of the service is to bring to the attention of the nation those patented inventions under which the owners are willing to grant licenses. It is hoped that such information will lead to greater employment opportunities in the reconversion period as well as permit industry to become acquainted with what is being done in various fields.

NEW MEANS of chemical analysis reported by General Electric involves shooting a beam of X-ray through an unknown chemical substance to see how much of the radiation is absorbed. This rapid method of identifying elements can be used with gases, liquids or solids and is expected to have extensive application in industry. X-rays, being intimately related to the inner atom, furnish a means of counting the number and indicating the kinds of atoms by which the rays are being absorbed. For a good first approximation, the count is unaffected by whether the atoms are free or have combined to form molecules of any sort or by whether the sample is hot, cold, solid, liquid or gaseous.

SOUNDPROOF ROOM so quiet that it approaches the quiet of the stratosphere is used by the Navy in its Brooklyn yard to test sensitive radio and electronic equipment. The room is mounted on 14 columns of rubber, each 16 inches in height, and is lined with deep layers of spun glass and other accoustical materials. Walls are nonparallel to reduce sound reflection.

SPOTWELDING applications to 100 assemblies used in making B-24 bombers have produced savings of 125 man-hours in working time per ship, a corresponding reduction in weight and a halving of original costs, according to records of Ford's Willow Run plant.

DEFECTIVES have been reduced and quality of 40-mm ammunition improved by using rubber crimping rings to seal the brass cases to the projectiles instead of rolling a crimped ridge. Hydraulic presses assemble the loaded cases and projectiles so that when pressure is removed the rubber returns to its original shape to effect the seal.

AIR WAR with Germany was won with planes and equipment which were not only designed but in production prior to Pearl Harbor. Even after 3½ years of fighting there was no effectual number of planes of later designs in combat, indicating the tremendous development necessary prior to quantity production. At the time of Pearl Harbor there were 1157 serviceable planes of which 526 were placed strategically to meet possible attacks on the Hawaiian and Philippine Islands. Within a few hours that number was reduced to 176.

MACHINE DESIGN

# How Odograph Plots Maps Automatically

By J. D. Faustman
aptain, Corps of Engineers
The Engineer Board,
Ft. Belvoir, Va.

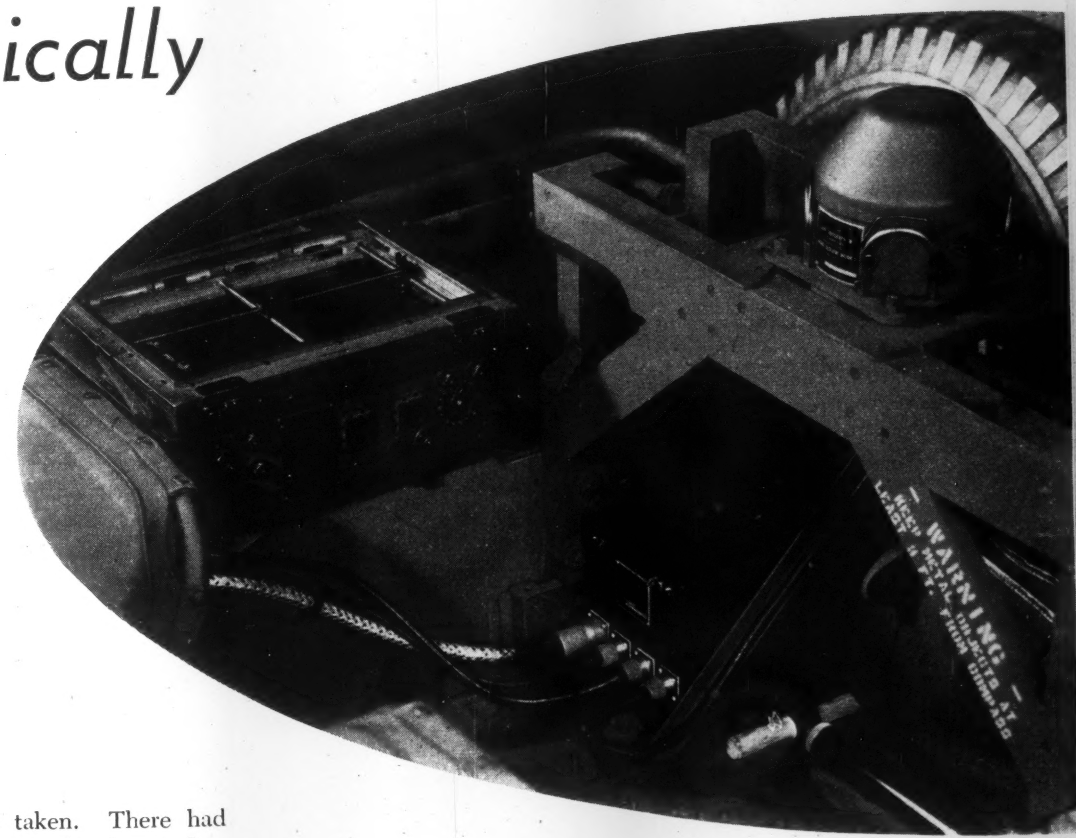


*Fig. 1—Map making instrument installed in a jeep for reconnaisance*

R EQUIREMENTS of an instrument for obtaining accurate and con- uous orientation of a mov- vehicle in modern war- e have been satisfied by e development of the odo- aph, an instrument that omatically plots the course taken. There had en many attempts in the past to develop such a ice for rapid route surveys but none had ad- nced beyond the planning and experimental stage. e Engineer Board therefore was directed by the ief of Engineers to undertake the basic development and research to produce an instru- nt suitable for military use.

Because of the large amount of research required, the Army requested the assistance of National Defense Research Committee, who turned the problem over to three organiza- s: The Department of Terrestrial Magnetism of the Carnegie Institution of Washing- D. C., the Monroe Calculating Machine Co., and the International Business Machines p. After an intensive cooperative research and test program, models were produced pos- ing the necessary reliability and accuracy. The odograph was then placed in quantity production and the first units were distributed to troops in August 1943.

Since the instrument automatically plots the course taken by a moving vehicle, unknown terrain is readily mapped by driving over it. In addition to mapping roads, the instrument may be used to plot locations of important points or objects either by driving the vehicle to the point or estimating its distance and azimuth from a known location of the vehicle with a range finder. In reconnaissance operations a map having the desired course can be placed by the plotting pencil can be watched to make sure the correct route is being followed.

When traveling to a given objective whose position is known relative to the starting point, it is possible to plot the objective on the odograph map table and then drive the vehicle in the general direction of the objective by following roads or other convenient routes. Since at any given time the relative locations of the objective and the vehicle are always shown on the odograph map, the vehicle can reach the objective even though the driver has no previous knowledge of the road net and terrain.

### Instrument Has Many Uses

Mapmaking and navigational functions of the odograph are often combined by using either the same or different instruments. Thus, maps made by one or more odographs can be used by other odograph-equipped vehicles. The unit is particularly valuable at night when guiding landmarks are not visible.

The odograph consists of three principal units:

1. Magnetic compass with an electronic follower system
2. Plotting unit with its 8½ x 11-inch map table, tracing pencil, and mileage counters
3. Power pack operating from the vehicle storage battery, interconnected by electrical cables and flexible shafts.

In *Fig.* 1 is shown the standard military installation in a jeep. The plotting unit is positioned for accessibility and the compass for optimum magnetic conditions. Having an infinitely variable type of map-scale gearing, the odograph can plot maps to any scale from 1:20,000 to 1:500,000. Mileage counters indicate the distance traveled east or west and north or south of a given starting point. The total miles traveled are also recorded. A means is incorporated whereby adjustments can be made so as to set in the magnetic declination, thus plotting on true north headings. The general size of the various components is observed in *Fig.* 1 and the weight of the entire installation is approximately 200 pounds.

### Requires Three Elements To Plot Course

In order to plot the course of a vehicle, it is necessary to know both the direction of travel and the distance traversed. Thus, a mechanical instrument to accomplish this function must contain: (a) A distance measuring element which is moved or turned by an amount proportional to
the actual distance traveled; (b) a compass element wh
provides a fixed reference direction; and (c) an integra
which provides for the resolution of the direction of
tion into components, one parallel to, and one perpend
ular to, the reference direction. It also provides for
summation (integration) of the distance components.
position of the pencil on the odograph map is controll
by two perpendicular movements, one of which is prop
tional to the sum of all the north and south components
motion of the vehicle, and the other to the sum of all

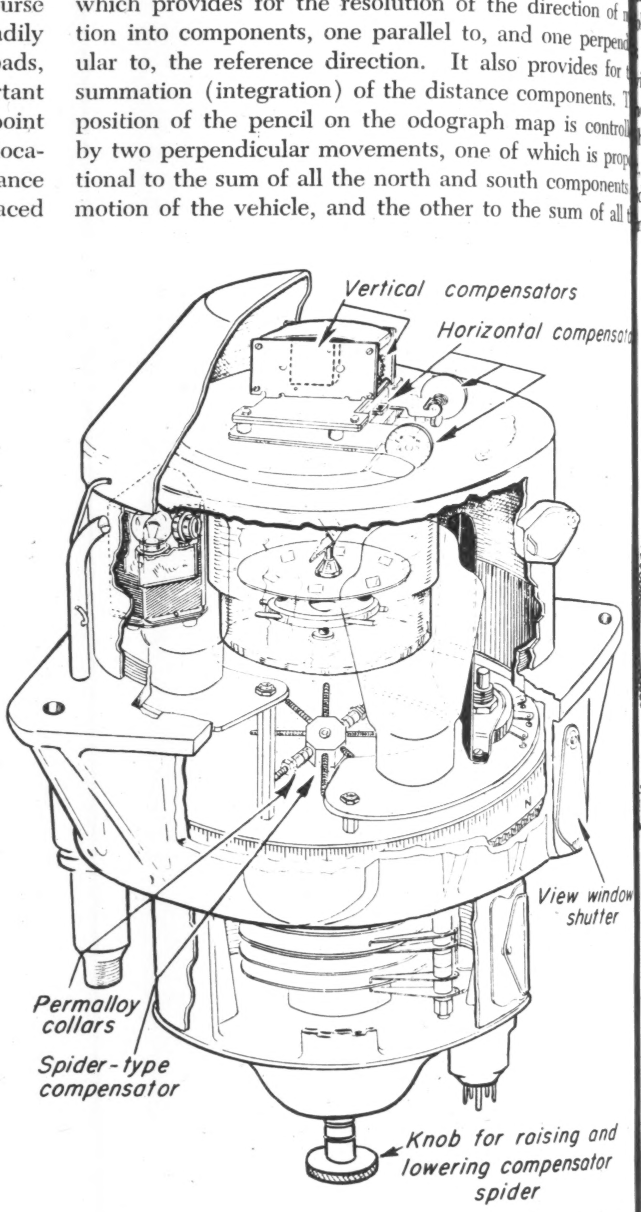


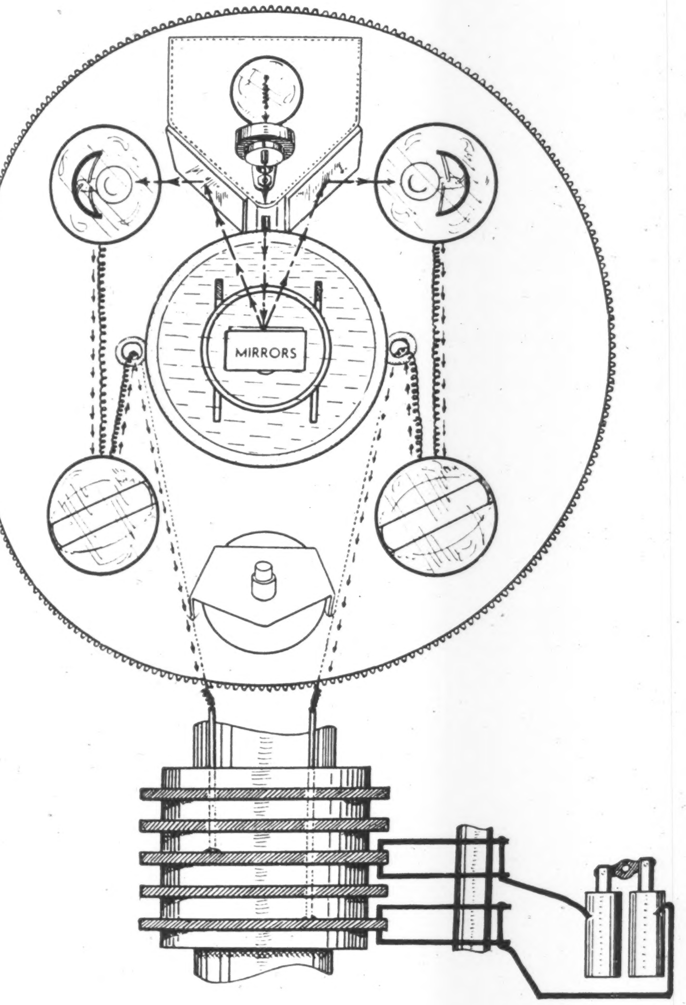
*Fig. 2—Odograph compass showing compensator system other features of design*

east and west components. This mechanical summat
is accomplished in the plotting unit.

COMPASS: The reference direction element is a ma
netic compass. This type of compass is used because
the relative complexity and size of a complete self-dire
ing or north-seeking gyroscopic compass, and because
shock and rapid changes in direction involved in land
erations are too severe for such apparatus. The effect
the vehicle's magnetic field upon the compass is co
pensated for by appropriately placed bar magnets a
permalloy collars as shown in *Fig.* 2.

So that the compass magnets may actuate the approp
ate mechanism in the plotting unit, the controlling e
ment is a beam of light projected into the compass bo
and then reflected from a mirror mounted at the center
the compass card, *Fig.* 3. Response of the compass ca

the influence of the earth's magnetic poles causes the ard mirrors to face in a northerly direction immediately llowing any displacement of the compass housing. Light reflected onto either of two phototubes. These, together ith the light source and two control thyratron tubes are ounted on a rotatable-ring assembly. After one of the ototubes has been activated by reflection of light upon current passes through the associated thyratron and one de of an electromagnetically operated clutch in the plotng unit, thereby causing the driving motor in that unit to



*ig. 3—Schematic view of odograph compass showing mirrors and phototube arrangement for electronic circuit*

tate a flexible shaft connected to the compass-ring asmbly until the reflected light beam strikes the other ototube. This action energizes the other side of the ectromagnetic clutch and causes a reversal in the direcn of rotation of the flexible shaft and the ring assembly. he result is a continual shifting of the clutch magnets and cillation of the rotatable ring, carrying the light source d phototubes back and forth so that, alternately, one ototube and then the other receives the light reflected m the compass mirror. The angle of rotation between e positions of reversal is approximately 10 degrees.

It is evident that, since the power is derived from the otor and associated magnetic clutches, the torque available to drive the compass rotatable-ring assembly carrying the phototubes (and also to drive any other connected mechanisms in the plotting unit) is limited only by the design of the mechanical system and does not in any way disturb the compass card itself. As the compass follower swings back and forth through its arc, a sine disk mechanism in the plotting unit controlling a Scotch yoke is driven by the motor through the same angles and is thus positioned according to the angle of azimuth of the vehicle.

Plotting Unit: The plotting unit must perform the mathematical operation of adding the components of $S \sin \alpha$ and $S \cos \alpha$, where $S$ is the distance traveled and $\alpha$ is the angle of azimuth of the vehicle's heading, with respect to the reference direction maintained by the compass. A convential Scotch yoke mechanism is used for obtaining a motion proportional to the sine and cosine of the angle of azimuth. Thus, if means are provided by which the rotary motion of the vehicle's odometer drive is variably coupled to the elements driving the plotting pencil with the degree of coupling proportional to the positions of the Scotch yoke, the motion of the plotting pencil then will be proportional to the amount of travel of the vehicle multiplied by the sine and by the cosine of the angle of azimuth of the vehicle. Position of the Scotch yokes is controlled by the compass.

## Utilizes a Positive Drive

Several mechanical and electrical principles which provide a degree of coupling between a driving and a driven unit proportional to a linear displacement were considered in the design of the integrating unit. Careful consideration was given to the effect which violent motion or shock might have on the mechaical elements involved in such an integrator and the decision was made that only a positive, mechanical system could be used.

In the integrating system the essential variable-coupling element is a pair of gears with stepped teeth. Length of the teeth along the face of the gears varies from a very short tooth to one running entirely across the face. Ends of the teeth form a one-turn spiral, as shown in *Fig.* 4. Two such gears are provided, rotating in opposite directions, with the amount of their rotation controlled by the odometer drive. These gears engage a driven gear which is moved parallel to the axis of the driving gears by the Scotch yoke. It is evident that, if this driven gear is in space between the two driving gears, it will not be turned. If a little to one side, it will be engaged by only the one longest tooth and will be rotated only one tooth for each revolution of the driving gear. As it is moved farther and farther toward the outside edge of the driving gear, the driven gear will be engaged by more and more teeth of the driving gear until, when it is at the extreme outer edge of the driving gear, it will be engaged by teeth all the way around and will be rotated by a maximum amount. If the driven gear crosses the central space between the two and begins to be engaged by the oppositely-rotating driving gear, a similar coupling will take place but with the driven gear rotating in the opposite direction. Thus, the amount of rotation of the driven gear will be directly proportional to its position along the axis of the stepped driving gears, and will fulfill the condition that its rotation is proportional to the sine of the angle of heading. A similar set of gears engaging a second driven gear, controlled by a second Scotch yoke gives motion which is proportional to the cosine of the angle of heading. The two driven gears are coupled to lead screws giving the two coordinates of motion to the mapping pencil, so that the motion of these screws and of the pencil is proportional, respectively, to the sine and cosine of the angle of heading.

Degree of coupling between the driving and driven gears varies by discrete steps, as the lateral motion of the driven gears picks up and drops one tooth of the driving gear. However, on account of an oscillation of the compass follower, the Scotch yoke continually moves back and forth by a small amount. The effect of this oscillation is to interpolate accurately between the integral number of teeth of the driving gear, so that the average rotation of the driven gear is proportional to the whole number of teeth plus a fractional number, to give the proper fraction so that the correct rotation of the driven gear results. It is also to be noted that this oscillation of the compass serves to eliminate backlash in the flexible shaft and associated gearing connecting the compass and plotting unit. *Fig.* 5 shows the interior mechanisms of the Monroe plotting unit.

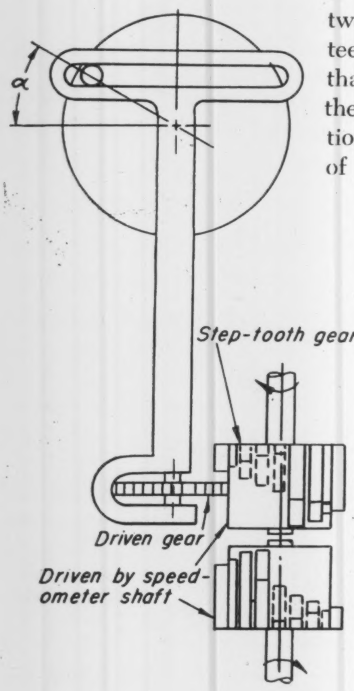


*Fig. 4 — Diagram of stepped-tooth variable coupling device of integrator*

### Variable-Speed Transmission Changes Scale

Scale of the map produced depends upon the various gear ratios of the odometer drive and of the coupling between the driven gears and the lead screws. To obtain a variable scale ratio, a set of gears is provided so that the coupling ratio between the odometer drive and the plotting pencil may be varied. The ratio-changing gears are much like a miniature of the variable-speed transmission of an automobile. By setting a dial controlling the position of these gears, the map may be drawn to any sca
between 1:20,000 and 1:500,000 as previously mentione
In military use of the instrument it is essential to dr
maps at different scales.

**Power Pack:** The power supply for the odograph is
conventional vibrator type power pack. Direct curre
at high voltage is obtained from the 6-volt vehicle batte
by converting the battery current to alternating current
means of the vibrator, stepping up the alternating curre
voltage through a transformer, and reconverting to dire
current by a rectifier tube. Current requirement for th
complete odograph instrument is 20 amperes at 6 vol
direct current.

### Sources of Error

There are several sources of error which reduce th
overall accuracy of the odograph, namely; direction err
(compass), distance error, slope-distance error, and m
chanical error.

**Direction Error:** The magnetism of a vehicle chang
when the vehicle is subjected to shock. Thus, in drivin
over rough terrain, the magnetism changes more than
smooth terrain, and a larger error is to be expected. Roug
terrain disturbs the dynamic balance of the compass, wi
some lowering in accuracy because of the fluctuations
the compass card. Compass errors are the major sour
of error in the odograph, and may amount to 2 per cent
more.

**Distance Error:** Distance input is taken off th
speedometer (odometer) drive gear in the vehicle tran
mission. The distance input will be most accurate wh
operating on hard smooth surfaces, with standard tires i
flated at the recommended pressure. On loose or roug
surfaces, some slippage will occur. Amount of slippa
depends upon several factors, and fixed rules covering th
many conditions that may be encountered cannot be s
up. When a vehicle becomes stuck in soft sand or mu
the wheels may spin considerably before the vehicle ge
out. In such a case the odograph would be turned
until the vehicle is free. The distance input error m
amount to as much as 2 per cent under bad conditions
slippage.

**Slope-Distance Error:** The odograph records the a
tual distance traveled over the ground. In terrain wi
many hills, the odograph will show a slightly greater mil
age than the distance scaled from a map, as the map is
projection of the terrain. It has been found that this err
can be regarded as negligible, unless an unusual numb
of steep up-and-down grades are encountered. For e
ample, on a 10 per cent grade the instrument will have a
per cent error.

**Mechanical Error:** A certain amount of play and lo
motion exists in the mechanical gear system. Close ma
ufacturing tolerances hold these errors to .5 per cent.

Because of the variable nature of the several sources
error, particularly changing magnetism and wheel sl
page, the direction of the error on a given run cannot
predicted. The magnitude of the error, however, can
predicted generally from a consideration of the type

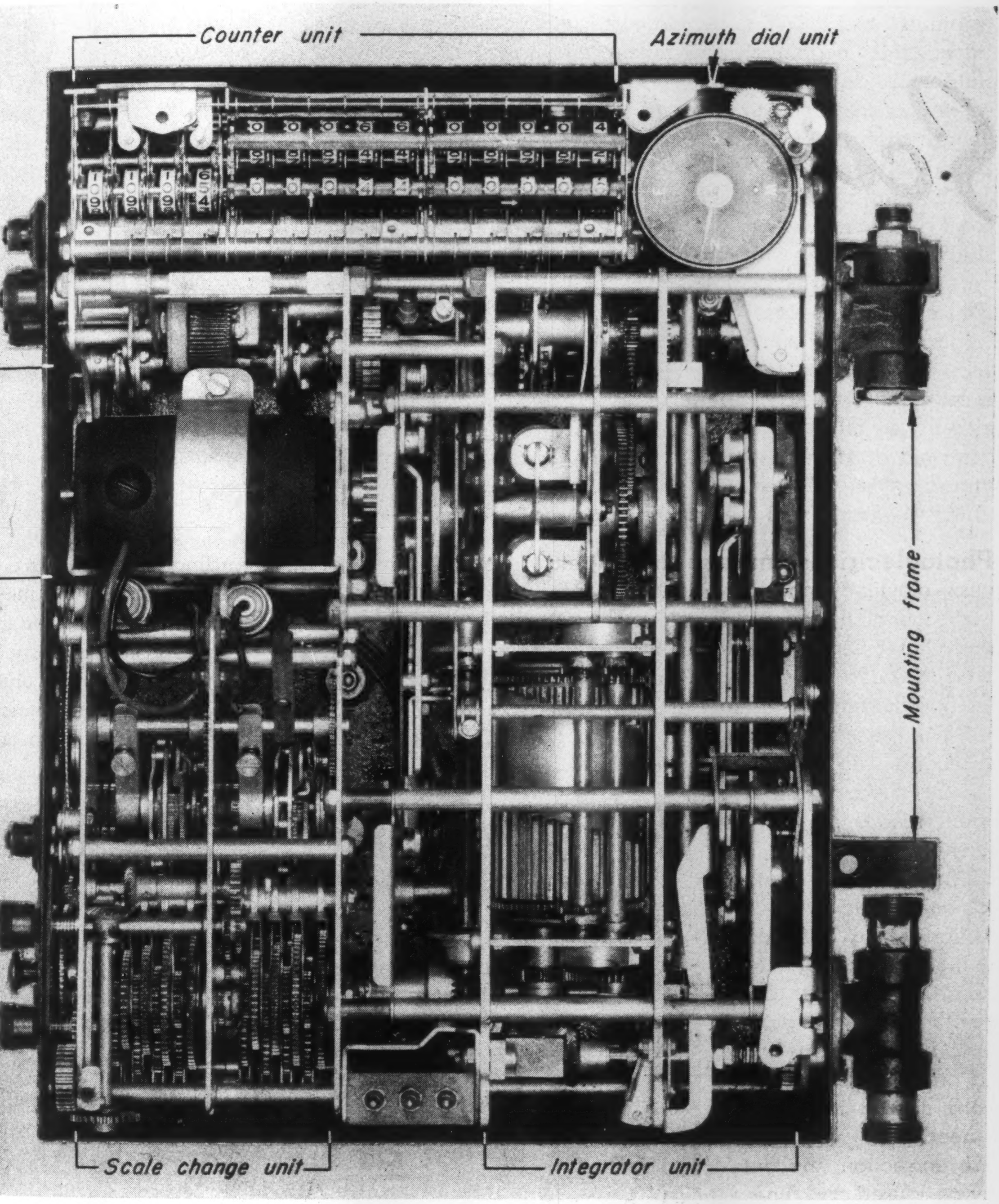


*Fig.5—Odograph plotting unit. Interior mechanisms are exposed by removal of case and map table*

rain to be traversed. On roads, the errors of closure ll average between 1 in 75 and 1 in 200, whereas on gh, cross-country terrain the error will average be-een 1 in 35 and 1 in 75.

### Has Postwar Possibilities

From a consideration of the function of the odograph, gether with a study of its operating characteristics and uracy, several postwar applications are indicated. In se applications a comparison with present methods and hniques reveals that with the odograph there will be a siderable saving of time, with resultant economies, and in most cases a higher degree of accuracy. Since it is basically a reconnaissance instrument, the odograph will find its greatest uses in surveys and mapping.

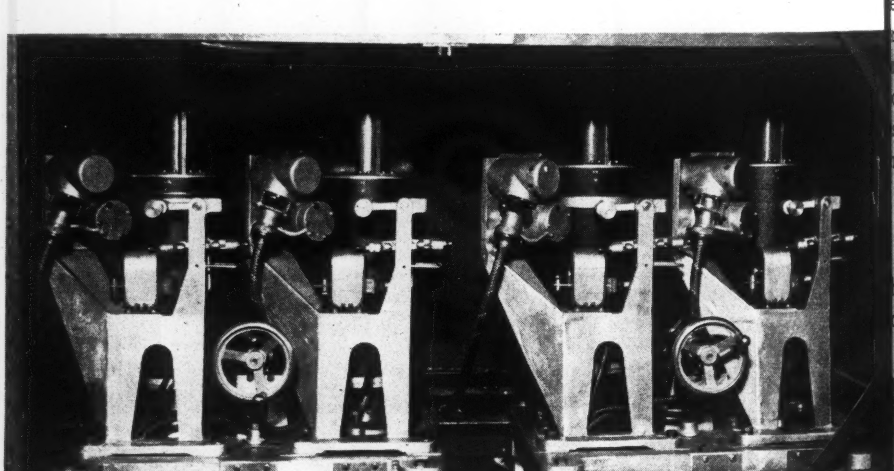
This odograph equipment will have particular value in all type of route surveys for highways, transmission lines, and similar projects. It is admirably adapted to geophysical surveys for oil and gas and for various types of geological explorations. Some of the military uses of the instrument will be applicable in postwar operations. For example, the navigation and dead reckoning features could be utilized for oil-well explorations in barren areas such as Arabia, and for oil and mineral explorations in roadless areas in Canada and Alaska.

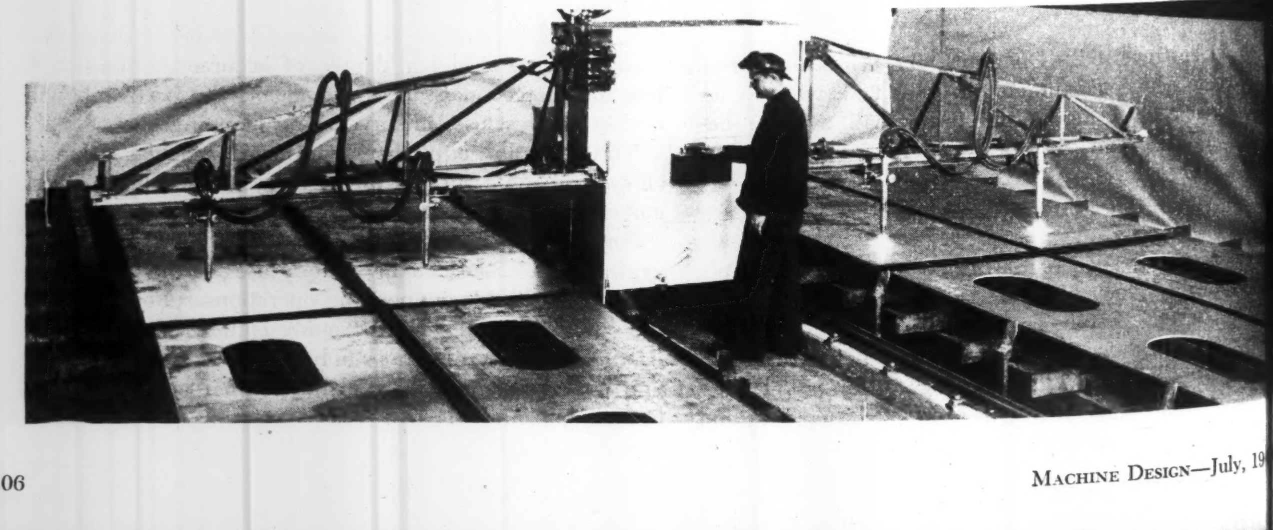
# Scanning THE FIELD for Ideas

**Photoelectric scanner** shown below, employing plastic-cylinder records, controls the flame cutting machine, below, developed by the ISC Flame Cutter Corp. Four master records are employed to guide the contouring motion of the machine. Operating in pairs for longitudinal and cross feed, one for distance and one for direction, the records are suitable for being filed as a permanent record or for future use. Each pair of records controls a motor feed associated with its direction of travel. Control markings on the records are small white dots for actuating a phototube pickup from reflected light of a projector lamp. If a record should no longer be needed or should require correction, the dots may be erased and a new coating of dots applied.

Track of the recording is helical like a dictating m chine record. As the drums turn they move ver cally to keep the consecutive dots in the path of t scanner unit. Several shapes may be combin on one set of records in the control unit.

This control is mounted in the machine, movi with it. For this reason, cutting in any location



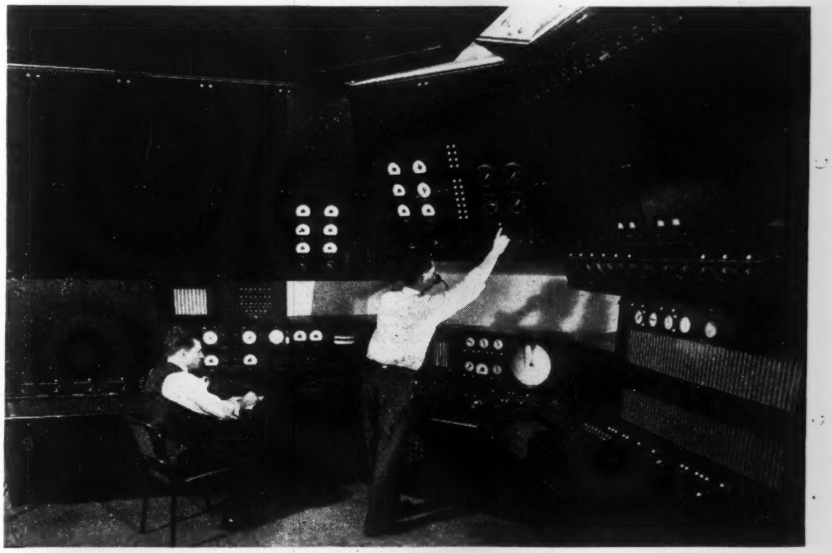


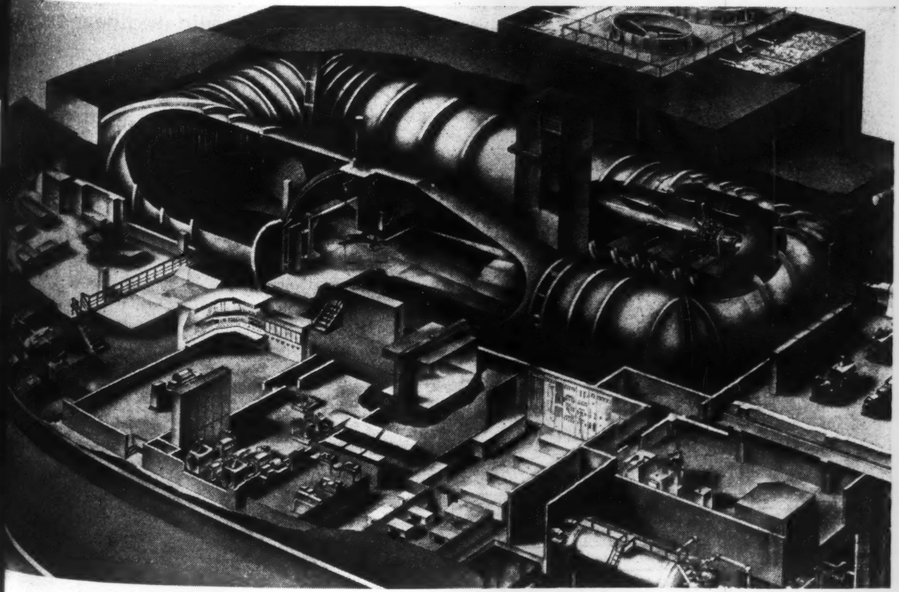
ssible and no alignment between
late and a template is required.
he records control fully all the mo-
ons of the machine during auto-
atic operation. When a cut is
arted, the torches are moved by
anual pushbutton to starting posi-
on. Then the automatic control is
t and the torches will follow a path
escribed by the four records. Both
rections of torch travel are moved
sitively by rack and gear. The
cord drums are geared to these
ck movements so that both torch
avel and record travel are inter-
nnected. Mirror contours are pro-
uced readily by reversing the di-
ection of motor travel with respect
record movement.

Records are prepared by reversal
the procedure for cutting con-
urs. A full-size drawing is placed
n the bed and the operator follows
e contours by pushbutton control.
s he does so a recording is made
the actions of the two drive mo-
rs. The operation may be simpli-
ed by recording important refer-
nce points and interpolating with
n auxiliary machine for this pur-
ose. Also scale drawings may be
sed by changing the gear relation-
ip between scanner and torch.

**Variable-density** wind tunnel
own in cutaway view, below, is
pable of testing airplanes in the 740 miles-an-hour range or the speed of jet propulsion and at pressure conditions equal to altitudes of 35,000 feet. Built by the Curtiss-Wright Corp., the design was developed jointly with the California Institute of Technology. Containing an air volume of 210,000 cubic feet, the tunnel may be used to test models of all types of planes, work formerly requiring three or four tunnels.

Length of tunnel is 178 feet, width 81 feet and maximum height 36 feet. Drive motors are 14,000 horsepower, 4800 volts and power service is adequate for 24-hour operation. Two 16-blade, 22-foot fans working in tandem produce the wind for test purposes. Employing four special compressors, air in the ¾-inch shell may be pumped to 60 pounds per square inch or four atmospheres. On the other end of the pressure range, the tunnel may be evacuated to 4 pounds per square inch, equivalent to pressures at 35,000 feet altitude. The test chamber may be "sealed off" from the remainder of the tunnel to permit changing the models without affecting the test pressure in the remainder of the tunnel. This precludes the possibility of workers getting the "bends" and saves as much as four hours normally required to pump maximum pressure.



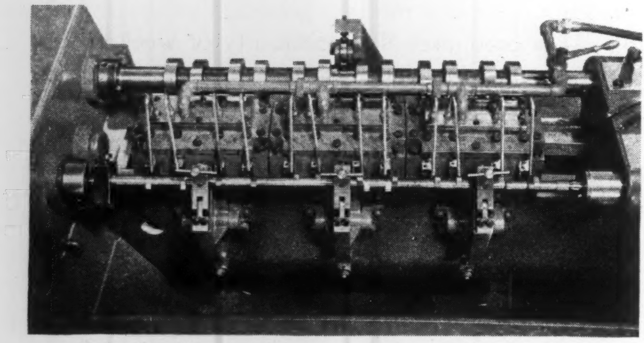


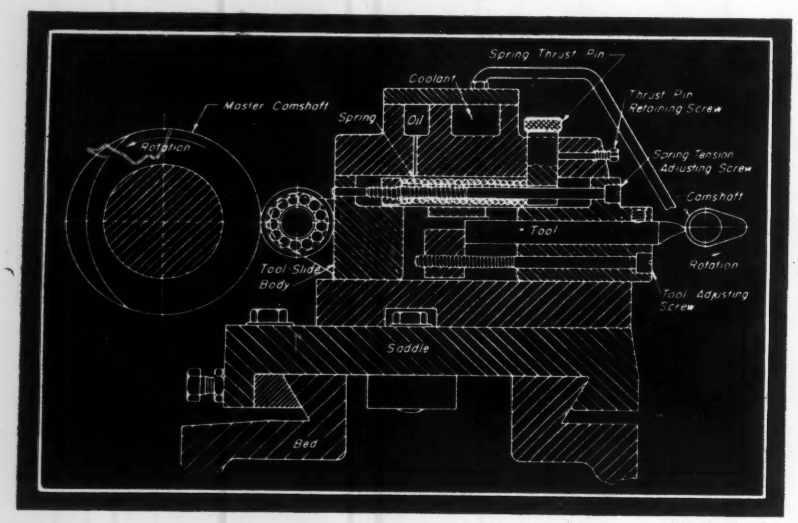
Test data are obtained from instruments shown above, in the control room adjacent the test chamber. Records of air-load forces are obtained simultaneously by pushing a button which causes the measured forces to be computed on perforated cards. Entire operation of the tunnel may be performed by three men from this control board.

**Profile turning lathe,** below, differs fundamentally from the normal method of cam turning in that no in-feed is required while the tools are cutting. Also the profile of the special tools is such that it is not necessary to rock the tool boxes in order to maintain the correct cutting angle. The principle of "copy generating" is shown in the sectional view below. Designed by Arthur Scrivener Ltd., Birmingham, England, the lathe employs a master cam that rotates at the same speed as the work. Between the master and the work is a horizontal slide with a roller at one end and a tool at the other.

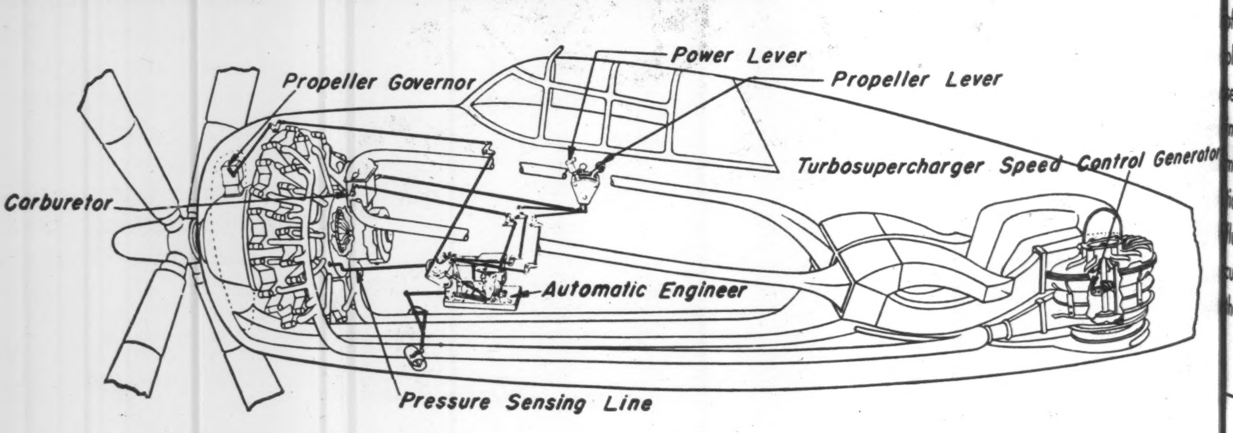
Radius of the tool point and roller follower have a definite relationship so that the form traced the tool will be the approximate shape for wh the master cam is designed. Cutting action of tool is accomplished by traversing axially along cam shaft. Action is similar to that of a knife t except for the generating effect of the tool tip. T longitudinal feed is hydromechanical, giving i nitely variable feeds with quick return.

For many automobile engine camshafts, a sin traverse of the tools produces a finished turned ca shaft of sufficient accuracy to obviate all subsequ operations excepting heat treatment and grindin

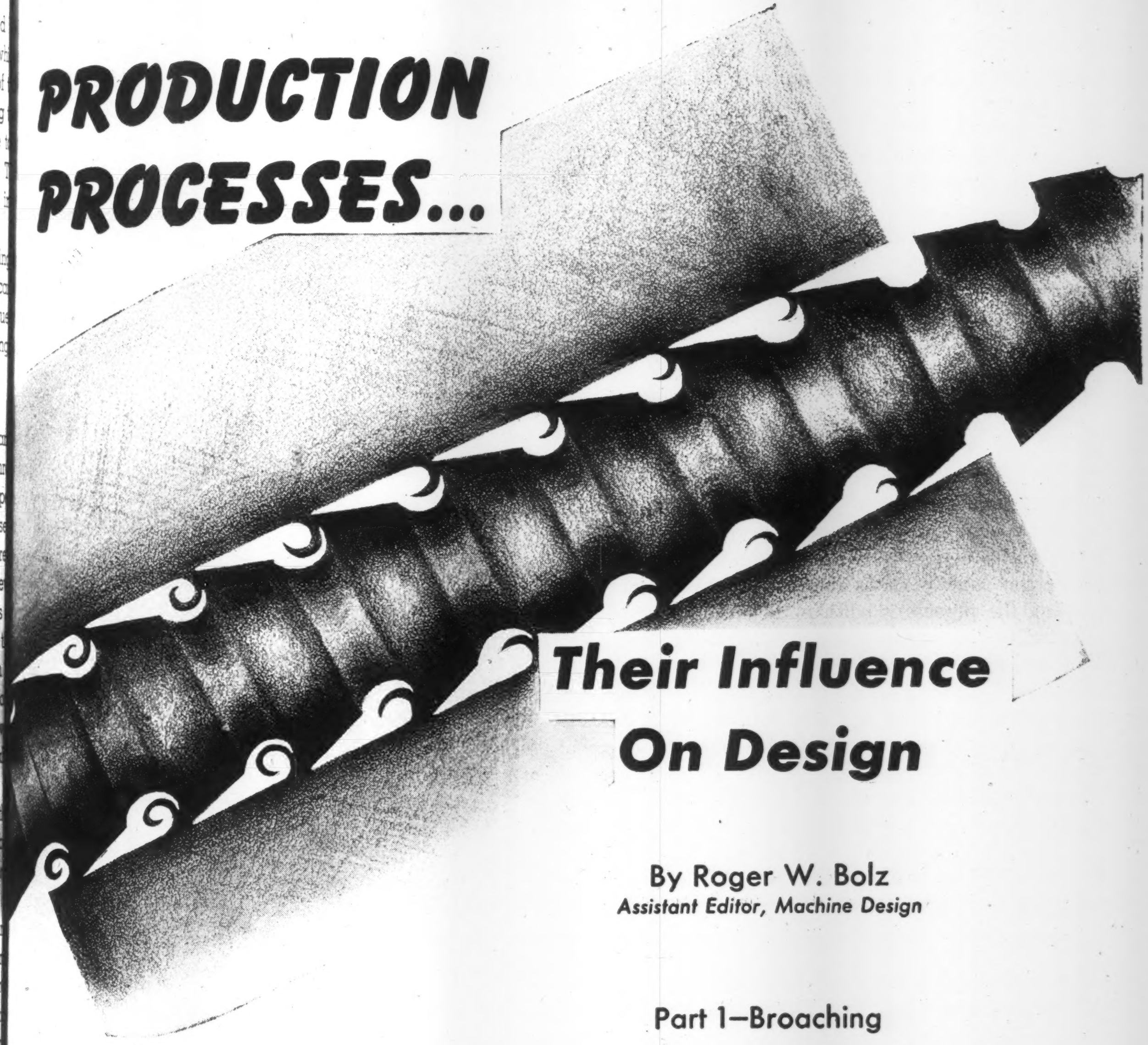



Spring Thrust Pin
Coolant
Thrust Pin Retaining Screw
Master Camshaft
Spring
Oil
Rotation
Spring Tension Adjusting Screw
Camshaft
Tool
Tool-Slide Body
Rotation
Tool Adjusting Screw
Saddle
Bed

**Automatic engineer** for aircraft power pla greatly simplifies the pilot's job by combining th important engine control jobs into one. Develop by the Army Air Forces and General Electric, it s the throttle, engine-speed control and manifold pre sure regulator to maintain the power output pre ously selected by the pilot regardless of changes altitude. Besides simplifying the pilot's job t unit, below, shown schematically installed in plane also reduces the human equation by emplo ing mechanical means to assure constant pow maintenance. At extremely high a tude or when necessary because battle damage, the unit also limits t maximum operating speed of the turb supercharger to a safe value, thus p venting damage to it and insuring continued performance,. By reduci the amount of control linkage a cables that otherwise would be quired this control reduces lost moti and provides more positive, faster a smoother response from the pow plant. When necessary during lo flights the pilot may take control aw from the automatic engineer and reg late manifold pressure and engi speed separately to improve fuel eco omy. Also "war emergency power" provided by the automatic control.


Power Lever
Propeller Governor
Propeller Lever
Turbosupercharger Speed Control Generator
Carburetor
Automatic Engineer
Pressure Sensing Line

# PRODUCTION PROCESSES...



# Their Influence On Design

By Roger W. Bolz
*Assistant Editor, Machine Design*

## Part 1—Broaching

***RODUCTION DESIGN which s economical and sound is ased to a great extent on a horough working knowledge f the various processes emloyed in manufacture. In a eries of articles on the most mportant mass production ethods, fundamental operaions and key factors which inluence design will be disussed from the standpoint of he engineer engaged in the design of machines.***

RECOGNIZED principally for its value as a production process suitable for manufacture at extremely high rates of output, broaching is now being applied to advantage in nominal-volume production work. Whenever the quantity of parts to be run is of fairly good size, say 2500 to 5000 pieces, the main benefits of broaching—minimum production cost per piece and high repetitive accuracy—become available. In cases of extremely exacting tolerances where no other process will produce the required results, even small-volume lots can be economical.

Besides the internal surfaces normally finished by this method, including holes of circular, square or irregular section, keyways, and multiple-spline holes, other surfaces such as teeth of gears, teeth of roller-chain sprockets and flat, curved or irregular external contours of practically any shape can be machined. Single and multiple cuts can be made both externally and internally with accurate dimensional relationship to one another at one pass of the tool. However, owing to the necessity for each tooth gullet of a broach to retain the chip created until the entire pass across a work piece is completed, the process is not particularly suited to parts requiring the removal of large amounts of material. The stock removed per tooth and length of the broach must be economically balanced to achieve optimum results. Great versatility can often be provided by arranging for an infeed fixture and repetitive cuts of the tool as shown in *Fig.* 1, but naturally the production rate is reduced accordingly and must be taken into consideration.

Broaching combines not only roughing and finishing operations, but as a rule removes stock to precision limits faster than any other known method producing similar results. In addition, sizing and burnishing teeth may be included on the broach to make possible even greater accuracy and the maximum in finish quality. Finish and dimensional accuracy are not affected by the accumulation of chips in the clamp and fixtures, a problem often encountered in machining operations.

Cutting speed or travel of the broach across the work is usually 15 to 25 times faster than that possible in milling the same piece. Hence steel forgings normally are broached at a speed of 25 feet per minute or more. Castings are often broached at speeds as high as 40 feet per minute, yet even at these high rates actual tooth travel through the material is approximately half that normally encountered with most production milling cutters. The lower tooth speeds and short cutting period create a minimum of abrasive action, consequently heating of either the broach or the work piece is minimized. Work temperature during the process is sufficiently uniform so that errors created by heating are seldom encountered.

Idle time between broaching cuts is kept at a bare minimum by rapid return passage of the tool. Loading, unloading and clamping in suitable fixtures is usually extremely simple. This is possible since the main force of the tool is in one direction only. Secondary forces created by this travel tend to separate the tool and work. Where the broach teeth have a shear angle a side force may be created. Actually, on most broaching fixtures, it is found necessary only to contain the work piece so that it cannot move; preloaded clamping, in the usual sense, is not always required and some parts can be worked without any clamping arrangement on the positioning jig.

Materials which require the maximum in effective a plication of lubricant to assure good surface finish can worked most successfully by broaching. Lubricant can directed and held where it is most effective. It actua flows between the tooth spaces under a slight pressu

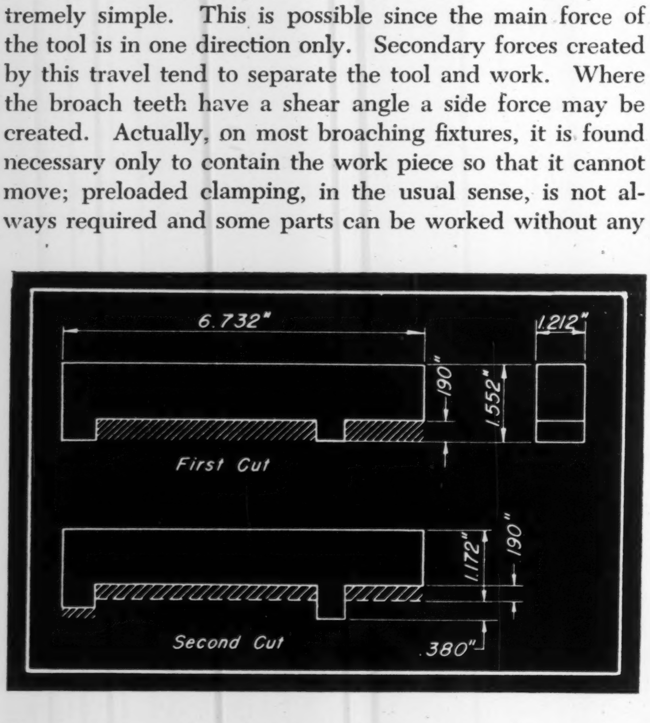
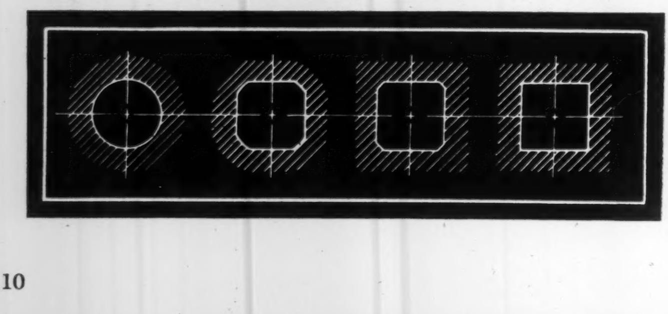


***Fig. 1—Above—Winchester carbine receiver broached in two passes removing .19-inch of material at each pass***

***Fig. 2—Below—Method of generating a square hole***



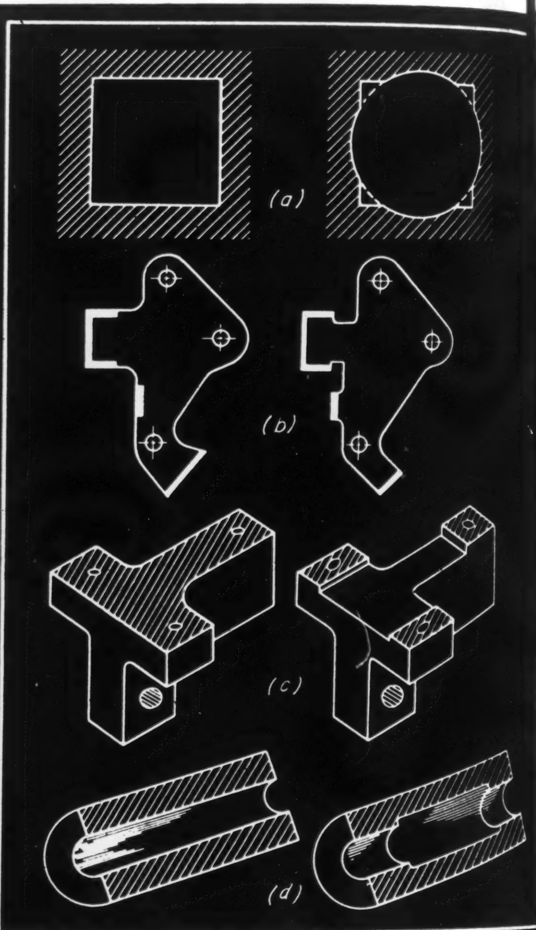
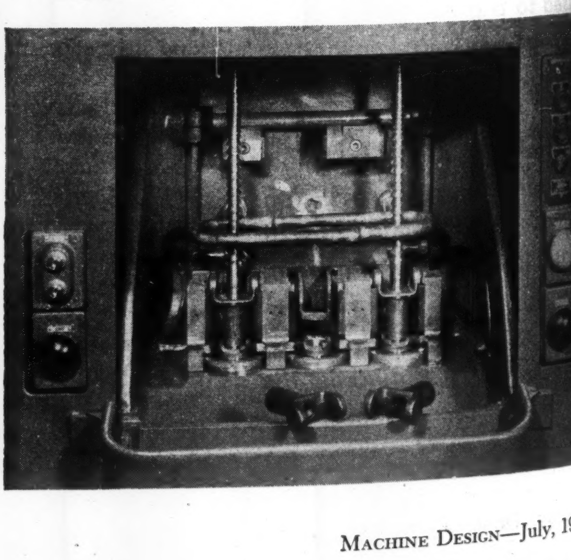


***Fig. 3—Above—Redesign can improve finish and a curacy as well as increase production. Designs show at the right are indicative of such improvements***

***Fig. 4—Below—Steel yoke with a straight splined ho being generated on a dual-slide vertical type machine***



lping to separate and lubricate the curling chips. Where terials tend to adhere to the cutting teeth, pressures as h as 900 pounds per square inch have been used to ce the coolant along the broach.

Unlike turning, milling, or shaping tools on which the st favorable operating conditions can be secured by rely adjusting the feed, a broach has a predetermined d or cut which is represented by the rise in the height each successive tooth. This, of course, is usually established by the broach designer but also must be recognized designing for broaching in order to adapt parts effecly to the process.

In single-pass broaching work it is not often economical remove more than ½-inch of stock, the volume of mate being limited by the capacity of the machine on which work is to be done. Where amounts exceeding the re val capacity or stroke of the machine are necessary, petitive or multiple-stroke cuts must be made. The to amount of material which can be removed for each ailable ton of ram pressure per foot of stroke is approxiately .06 cubic inch for cast iron, .04 cubic inch for mem-hard steel, and .03 cubic inch for tough alloy steels ving a rockwell C hardness over 28. In TABLE I the relionship between tooth pitch and chip area is correlated the various common pitches used in broaching. Since ese chip area figures represent the product of chip thicks and length of the cut, the maximum stock removal any size part can be approximated for any machine ailable.

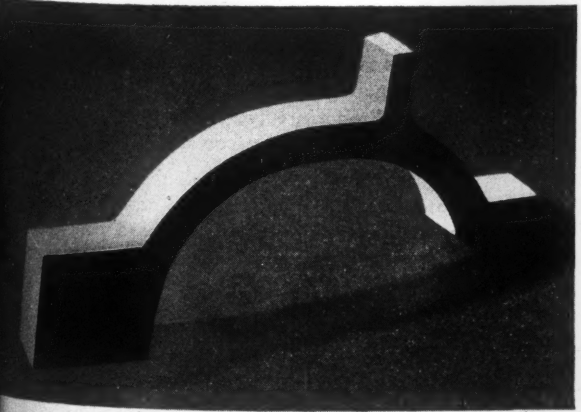
ROUND PULL BROACHING: This method is used for procing holes of high accuracy and fine finish. Holes may finish broached from drilled, reamed, bored, cored, nched, or hot-pierced holes. Starting hole size should 1/32-inch smaller than the finished hole for small diamers, increasing to as much as 1/16-inch smaller on large ameters. To assure clean-up and good finish, the allowce on diameter should never be less than 1/64-inch. here cored holes are planned, draft angle and size variaons must be taken into consideration to assure entry of e pilot and good clean-up. If extremely smooth finish is ecessary, burnishing teeth can be used at the end of the ol. However, these are recommended only for cast iron d nonferrous materials.

ROUND PUSH BROACHING: Employed to produce the me results as pull broaching, this method usually is restricted to parts requiring a relatively small amount of stock removal, since the tool must be comparatively short in order to have adequate stiffness. Push broaches are used for sizing holes after drilling, reaming or pull broaching, and for smoothing of die-cast and similarly produced holes. Special sizing broaches can be used for removing scale and correcting distortion in the holes of hardened gears, etc.

BURNISHING: A cold-working operation, burnishing removes no metal but only compresses and smoothes out fine surface irregularities. Contraction usually follows the expansion produced by burnishing. Amount of contraction varies with the material, size of the hole and construction of the part and can be determined most easily by experiment. Only soft, ductile materials are well suited to the burnishing operation.

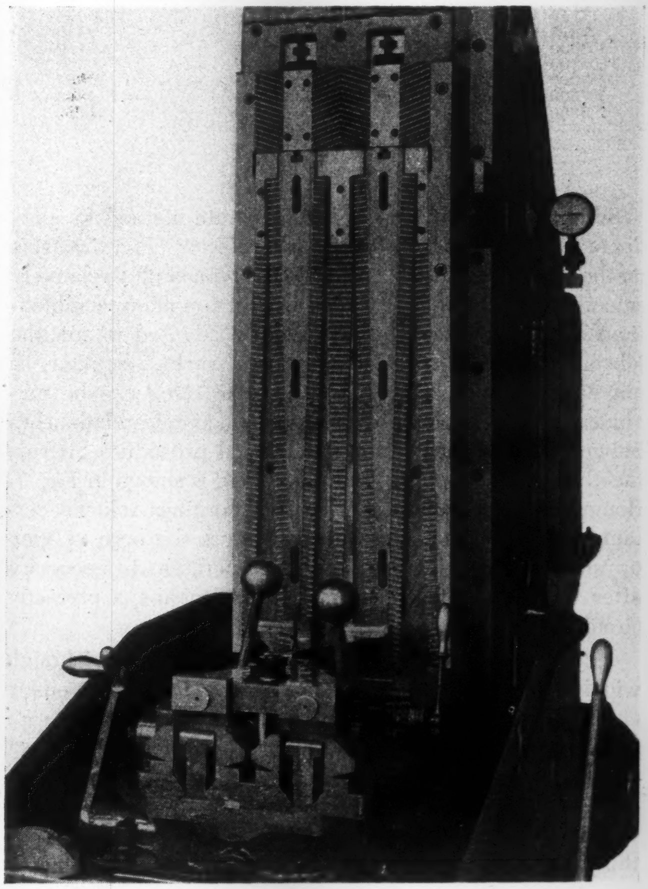
SQUARE, RECTANGULAR, AND HEXAGON HOLE BROACHING: This process is similar to round push and pull broaching. Such internal shapes usually are started from drilled, punched or cored holes. The method of generation is illustrated in *Fig. 2*. However, where possible, an oversize starting hole should be used as shown in *Fig. 3a*. This makes possible a shorter, more economical tool and much faster production.

MULTIPLE-SPLINE BROACHING: Splines of any required shape can be generated by broaching. Both straight (*Fig. 4*) and spiral splines are produced having involute, straight, angular-sided, radius lobe, and inverted shapes. As a rule the work is first drilled, bored, or reamed to



*5—Above—The entire contour of this irregularly shaped machine part was produced in two passes*

*Fig. 6—Below—Two bearing caps are produced at one pass on this progressive type broaching arrangement*



proper minor diameter and then broached, cutting the splines only. Where desirable, the hole can be prepared as for round broaching and a combination round-and-spline broach employed to finish the minor diameter as well as the splines. This type tool is particularly useful on high-production jobs requiring extreme accuracy. Present practice is to finish the minor diameter with the broaching tool when producing involute splines.

Spiral spline broaches generate tooth shape in much the same manner as straight spline broaches. Where the helix angle of the spiral is less than 8 degrees it is possible to pull the tool through the part with a ball bearing puller, the broach itself revolving to generate the proper helix. If the angle is larger than 8 degrees, a lead screw is necessary to assure positive and accurate generation.

Involute spline or internal gear broaches can be made to produce any number of teeth and are used for producing gears with as small as 48 diametral pitch teeth. In these gears the involute form is generated and checked from the base diameter. For extremely accurate parts or those heat treated after the first cut, a sizing or finishing operation should be used.

TABLE I*

**Relationship of Tooth Pitch and Chip Area per Tooth (chip thickness times length)**

| Pitch of Broach Teeth (inches) | Chip Area for Steel (sq in.) | Chip Area for Cast Iron (sq in.) |
|---|---|---|
| .2500 | .0014 | .0028 |
| .3125 | .0022 | .0044 |
| .3750 | .0031 | .0062 |
| .4375 | .0042 | .0084 |
| .5000 | .0055 | .0110 |
| .5625 | .0070 | .0140 |
| .6250 | .0086 | .0172 |
| .6875 | .0104 | .0208 |
| .7500 | .0124 | .0248 |
| .8125 | .0145 | .0290 |
| .8750 | .0168 | .0336 |
| .9375 | .0193 | .0386 |
| 1.0000 | .0220 | .0440 |
| 1.2500 | .0330 | .0660 |
| 1.5000 | .0495 | .0950 |
| 1.7500 | .0673 | .1346 |
| 2.0000 | .0880 | .1760 |

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* Cincinnati Milling Machine Co.

**SURFACE BROACHING:** A fast, accurate method for producing flat, curved or irregular surfaces (*Fig.* 5), this method of broaching is unexcelled for making a relatively smooth, even finish at low cost. One of the many combinations of tools available usually can be adapted to suit the job on hand, *Fig.* 6. Gear segments, racks, serrations of any variety, slots, teeth, curved guides, etc., can be machined at one pass with exceptionally accurate relationship between the various surfaces. Typical procedure for surface broaching a connecting-rod forging is shown in *Fig.* 7. Removal of the surface skin on such forgings relieves certain internal stresses which in turn cause the bore to open up slightly. A truing and boring operation is necessary after broaching such a cap and rod to assure a perfectly circular bearing seat.

**KEYWAY, SLOTTING AND CUTOFF BROACHING:** Though widely used to produce accurate keyways to extremely close tolerances, a keyway-type broach is also adaptable for sawing or cutoff work, especially where very narrow or sinuous key slots are required to exacting dimensions, *Fig.* 8. Effectiveness of coolant action with this type tool obviates heat distortion troubles otherwise present in such thin-bladed tools. Rise per tooth usually is restricted to a maximum of .0027-inch to avoid marking or scoring of t slot sides by the spreading action of chips formed wi heavier cuts. This limits the depth of slot to about 7/1 inch on a 66-inch capacity standard machine. Slots ⅝ ¾-inch deep are possible on 120-inch machines.

**DESIGN:** Decision to produce a part by broachin should be followed by a critical analysis of the part desig to assure lowest cost and maximum ease of productio Blind or extremely long holes should be avoided. Broac ing of blind holes requires a stripper broach having bu nishing buttons to open the hole slightly in order to pre vent the broach teeth from dragging on the finished su face as it is withdrawn. These tools are not satisfacto on high-production work. Long holes, of course, woul require an extremely long broach and possibly a set several. Such holes should be chambered as shown i *Fig.* 3*d* both to improve accuracy and finish as well as t reduce the cost. Similar consideration of large flat su face-broached areas is of importance. Where possibl such areas should be reduced to a series of bosses as i *Fig.* 3*c* to simplify tooling and shorten the cutting cycle.

Where sharp corners can be avoided, again the toolin and the production time can be improved. Use of a relie or undercut simplifies the design, *Fig.* 3*b*. If blended co toured surfaces, *Fig.* 5, are required, a combination broac can be arranged to handle the job properly.

For slotted parts such as those shown in *Fig.* 12, eithe sharp corners or radii are equally satisfactory at the bo tom of the slots. The broaching tool in either case ca easily be sharpened. However, chamfered corners rathe than rounded ones should be specified for the outer edge or corners. A chamfer at the outer edge will reduce th cost of the broaching insert required and make an almos impossible sharpening problem a simple one.

It is advisable at all times to maintain uniform wall sec tions on parts to be broached and avoid frail or thin sec tions. Sections should be at least sufficient to withstan the fixture retaining pressures without distortion. Whe no locating surfaces are to be machined before the broach ing operation, design of the part must be such that an piece can be easily located without severe misalignme

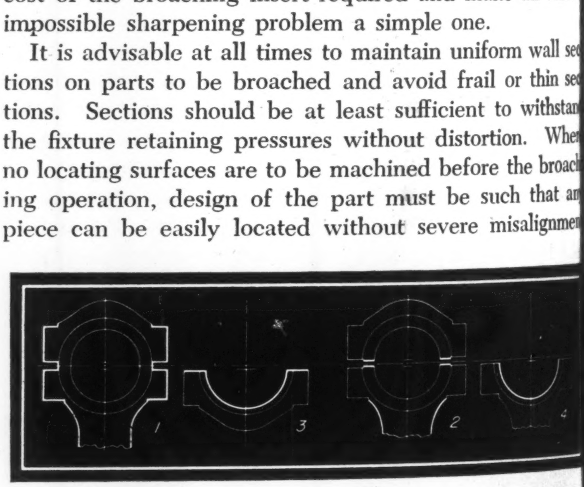
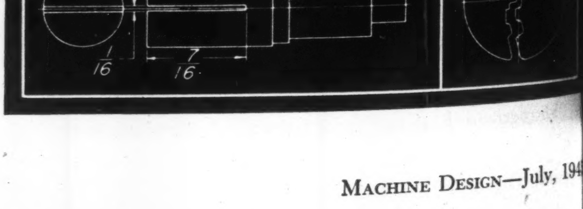


*Fig. 7—Above—Procedure and ayout of progressi stations for finishing a one-piece connecting-rod forgi*

*Fig. 8—Below—Exacting tolerances can be held whe narrow or sinuous slots such as these are required*



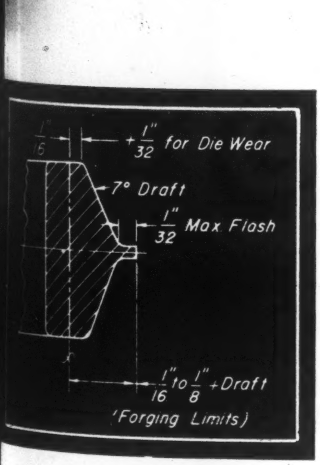
sulting from casting, forging or forming variations. With ng slender machine parts it often is desirable to "condion" the part by first broaching locating spots to assure raightness in the wide cuts which follow.

Forgings should be held as closely to limits as possible order to allow only a minimum of stock for broaching. general idea as to the allowances for a typical forged ction can be gained from the illustration in *Fig.* 9. On e other hand, castings require a greater stock allowance order to allow the broach to rough beneath the incluons, scale and hard spots. Cold-punched or pierced holes esent much the same problem as castings. Such stampgs that have been cold-worked or punched should be rmalized before broaching. Parts having a nonuniform ensity such as that found in porous forgings should be oided since bad tearing and drift of the broach often sult.

MATERIAL: Any material that can be machined can be roached. The higher the machinability rating, though, e more easily and economically it can be worked. For est results hardness should be held between rockwell C25 d C35. Harder materials can be worked but sometimes volve problems of wear and lubrication. Softer ones are ften too "mushy" or tend to tear, burr or adhere to the tting teeth and overload the tool.

Various materials and hardnesses sometimes require arying cutting speeds for fine finish, good tool life and aximum production. Softer nonferrous materials can be roached at much higher speeds than steels. Malleable d cast irons can be broached at speeds around 40 feet er minute and brass and aluminum at speeds as high as 0 feet per minute. However, maximum available mahine speed at present is limited to about 43 feet per minte and at this speed actual cutting time is so short that aster speeds, often accompanied by somewhat lowered ol life, are not justified. A maximum speed of 33 feet er minute usually is used for steels of normal hardness, ropping as low as 16 to 17 feet where a material hardess of rockwell C40 to 43 must be utilized. Stainless teels can be broached but present a definite problem here fine finish is necessary. Stainless parts are successully broached to close tolerances but surface finish as a ule is poor since the material tends to tear.

TOLERANCES: The close accuracy and uniformity obainable in surface broaching depends to a great extent pon heavy rigid fixtures and a solid machine slide having s little deflection as possible. In internal broaching accuracy is, to a great extent, more dependent upon uniformity of material. Diametral tolerances on holes and splines of plus or minus .0002-inch can be held on small holes where essential. Width of splines can be maintained at plus or minus .0002-inch, and spline spacing within plus or minus .0001-inch. These tolerances are affected by broach design and allowance for wear on the finishing



$\frac{1}{16}$
$+\frac{1}{32}$ for Die Wear
7° Draft
$\frac{1}{32}$ Max Flash
$\frac{1}{16}$ to $\frac{1}{8}$ +Draft
(Forging Limits)

*Fig. 9 — Left — A typical forged section showing usual limits necessary to obviate overloading on production runs*

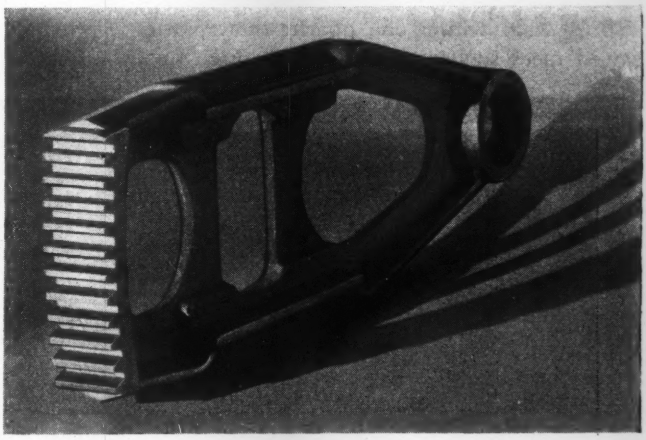


*Fig. 10—Above—Gear segment finished at one pass*

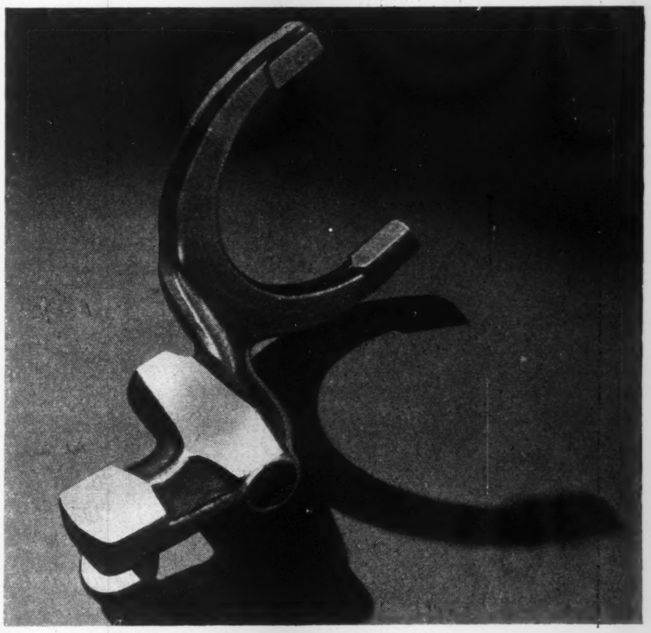
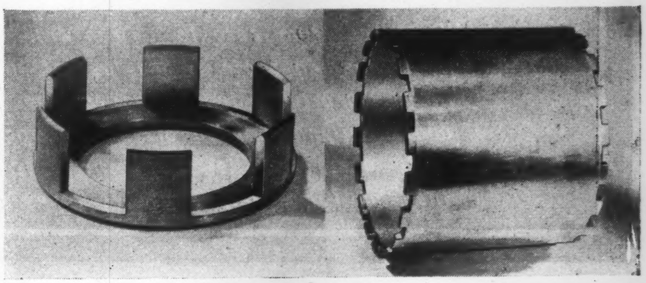


*Fig. 11—Above—Clutch yoke broached externally and slotted to exacting limits in a single stroke*

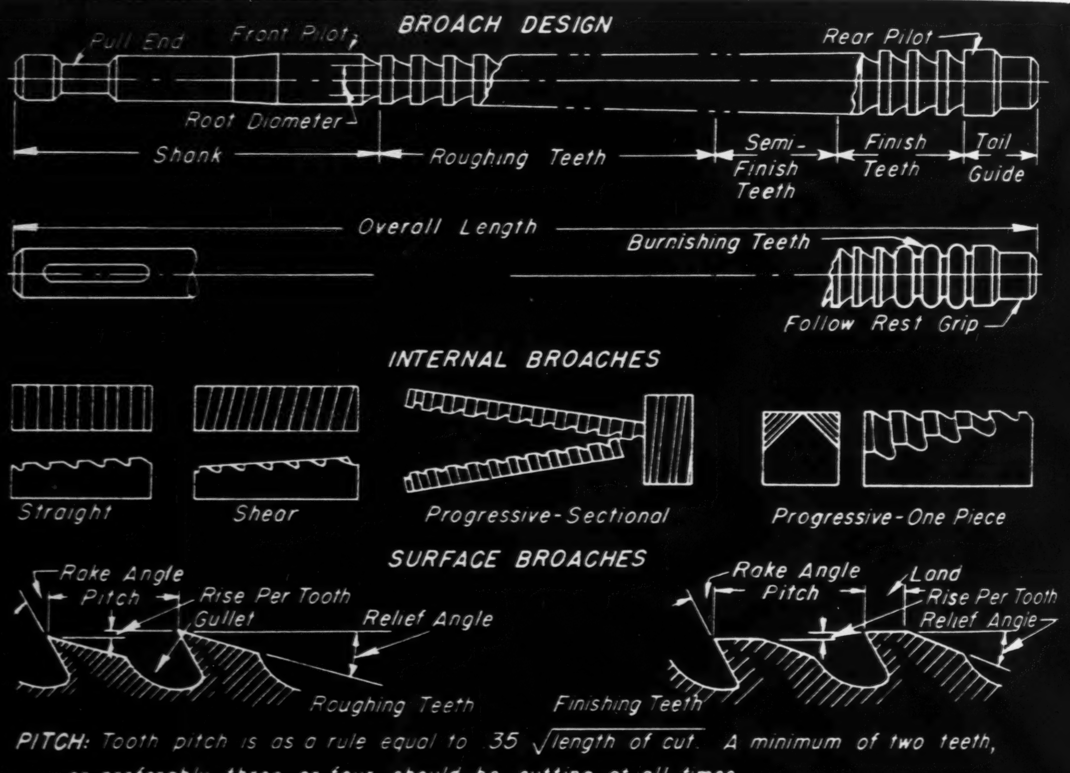
*Fig. 12—Below—Lugs formed by broaching across part, using an indexing fixture. Spacings accurate to .0003-inch*



teeth. External gears and racks, broached one tooth at a time, can be held to plus or minus .0005-inch on tooth spacing and form. On high-production runs where the entire gear segment or rack is cut in one pass as shown in *Fig.* 10, even closer tolerances can be held.

In surface work, straddle-broached faces similar to those shown in *Fig.* 11 are normally made to plus or minus .002-inch but can be held within one quarter-thousandth of an inch. Flatness of some surface-broached faces can also be held within one quarter-thousandth. Slotted or sawed faces such as that shown in *Fig.* 8 can be held to plus or minus .0002-inch, plus or minus .002-inch bei the normal economical limits. Likewise straddle-broach lugs on parts similar to that shown in *Fig.* 12 are held spacings accurate to .0003-inch where design requires.

Collaboration of the following organizations in the prepa tion of this article is acknowledged with much appreciatio Broaching Tool Institute; Cincinnati Milling Machine C (*Fig.* 1); Detroit Broach Co.; Foote-Burt Co. (*Figs.* 5, 6, 7, 1 11, 12); Illinois Tool Works; H. R. Krueger Co.; Lapoin Machine Tool Co.; National Broach & Machine Co.; The O gear Co. (*Fig.* 4).


BROACH DESIGN
Pull End
Front Pilot
Rear Pilot
Root Diameter
Shank
Roughing Teeth
Semi-Finish Teeth
Finish Teeth
Tail Guide
Overall Length
Burnishing Teeth
Follow Rest Grip
INTERNAL BROACHES
Straight
Shear
Progressive-Sectional
Progressive-One Piece
SURFACE BROACHES
Rake Angle
Pitch
Rise Per Tooth
Gullet
Relief Angle
Roughing Teeth
Finishing Teeth
Rake Angle
Pitch
Land
Rise Per Tooth
Relief Angle

**PITCH:** Tooth pitch is as a rule equal to $.35\sqrt{\text{length of cut}}$. A minimum of two teeth, or preferably three or four, should be cutting at all times.

**RISE PER TOOTH:** In free-cutting steel this is usually .004 to .006-inch on diameter for spline; .0015 to .003-inch on diameter for round; .001 to .005-inch for keyway and surface broaches. Rise per tooth as high as .015-inch can be used on trimming broaches where the cut is across the work and relatively narrow. Slotting broaches should not exceed .0025-inch per tooth. Values are often doubled for cast iron, malleable iron or brass. For burnishing and sizing, rise per tooth is not over .001-inch and often .0001 or less.

**RAKE ANGLE:** For cast iron the rake angle is usually 6 to 8 degrees positive; hard steel, 8 to 12 degrees positive; soft steel, 15 to 20 degrees positive; aluminum, approximately 10 degrees positive; brittle brass, 5 degrees negative to 5 positive.

**RELIEF ANGLE:** Normally this falls between $\frac{1}{2}$ and 3 degrees for internal broaches and up to $3\frac{1}{2}$ degrees for surface broaches.

**LAND:** Varies from zero to $\frac{1}{16}$-inch wide and usually is graduated in length from the first finishing tooth to the last on internal broaches. Often calculated as .2 times the pitch. The land varies from zero to .005-inch for surface broaches and usually is backed off at $\frac{1}{4}$-degree.

**GULLET:** Area of gullet space should be ten times the cross-sectional area of the chip for steels and five times for material such as cast iron.

# Causes of Failure in Heavy-Duty Bearings

**By L. M. Tichvinsky**
*Vice President in Charge of Engineering*
*American Bearing Corp.*

DIESEL-ENGINE bearings, because of the high cyclic loads to which they are subjected, perate with high maximum pressures which tend to hasten failures e to common bearing defects. This ticle discusses the types of failures ost often encountered in the critlly loaded bearings of a diesel, and dicates some preventive measures hich, it is hoped, will aid designers providing maximum bearing life in other types of machines.

Severest operating conditions are encountered in the pisn-pin, connecting-rod and main bearings, the principal charteristics of which are as follows:

PISTON-PIN BEARINGS: Maximum pressures are high—of e order of 6000 pounds per square inch—due to large peak rces in conjunction with comparatively small bearing diamters. Because of a swinging or rocking motion of the connectg rod, the load acts over a small arc of the bearing circumfernce, spanning approximately 30 degrees. Lubricating-oil temerature is rather high because the oil usually has passed through e main and connecting-rod bearings. Piston-pin surfaces are ard (approximately 60 rockwell C), often chromium-plated, nd smooth (below 10 microinches). The bearings are designed ith small clearances, and successful operation depends greatly pon the damping properties of the lubricating oil.

CONNECTING-ROD BEARINGS: Maximum pressures are high lso and may reach values of approximately 3000 pounds per quare inch. The upper shell supports most of the load in a 2-ycle engine, while in a 4-cycle engine both upper and lower earing shells carry considerable load. Connecting-rod bearings re better cooled than are main bearings as a result of the ventiation produced by the rotation of the crankpins. Lubricating l is supplied usually from the main bearing, being forced hrough the oil grooves and the inclined bores of the crankshaft. rankpin surface hardness varies from 200 to approximately 350 rinell and depends on the bearing-material hardness. Surface nish of the crankpins is made as mooth as practicable. These bearings re designed with clearance ratios arying from .0007 to .0015-inch per nch. Smaller values are used in 4-cycle diesel-engine bearings.

MAIN BEARINGS: Maximum pressures are moderate and may reach alues of approximately 1500 pounds er square inch. The lower bearing hell usually is more highly loaded han the upper. These bearings are bricated with cool lubricating oil irect from the heat exchanger and

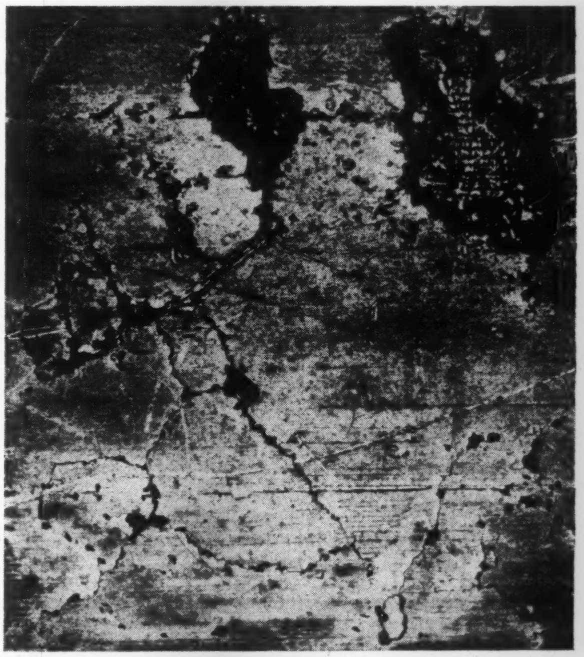


*Fig. 1—Fatigue failure of connecting-rod bearing from large V-type diesel engine after 400 hours of operation. Peak load produced a pressure of 3320 pounds per square inch in this thin Satco steel-backed bearing. Approximately 4¾ times full size*

TABLE I

**Composition of Heavy-Duty Bearing Materials, Per Cent**

| Material | Tin | Lead | Antimony | Copper | Cadmium | Silver | Nickel | Arsenic | Calcium | Aluminum | Zinc | Magnesium | Iron |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Cadmium-Silver SAE No. 180 | .01 | .02 |  | .50 | 98.0 | 1.00 |  |  |  |  | .02 |  |  |
| Cadmium-Nickel SAE No. 18 | .10 | .01 | .18 | .04 | 97.8 |  | 1.80 |  |  |  |  |  |  |
| Copper-Lead |  | 25.00 |  | 75.00 |  |  |  |  |  |  |  |  |  |
| Lead-Base Babbitt Navy Grade No. 6 | 5.00 | 80.00 | 14.00 | .50 |  |  |  | .20 |  |  |  |  |  |
| Satco | 1.00 | 98.00 |  |  |  |  |  |  | .50 | .07 |  | .08 | .01 |
| Tin-Base Babbitt Navy Grade No. 2 | 88.00 | .35 | 7.50 | 4.00 |  |  |  | .10 |  |  |  |  | .03 |

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From a paper presented at a recent meeting f the Cleveland section of the American Soety of Mechanical Engineers.

header. Main-journal surface hardness is the same as that of crankpins, and the bearings are designed with clearance ratios varying from .0008 to .0015-inch per inch.

Failures of diesel-engine-bearings may be caused by one or more of the following:

1. Fatigue of bearing metal under high cyclic loads
2. Excessive or insufficient hardness of the bearing material.
3. Corrosion of the bearing metal by the lubricating oil
4. Inadequate bond between bearing metal and bearing shell
5. Assembly errors
6. Foreign particles
7. Miscellaneous, such as defective manufacture, extrusion wiping, etc.

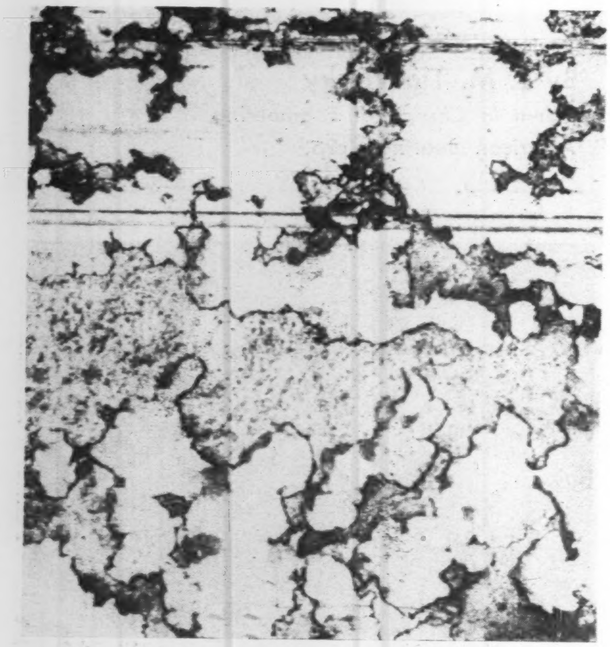
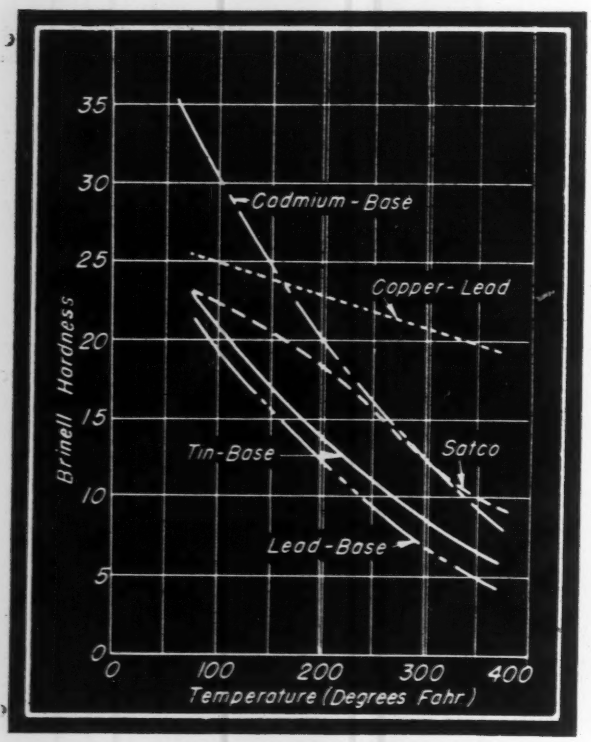


*Fig. 2—Above—Fatigue failure of trimetal bearing under similar operating conditions to Fig. 1. Approximately 7 times full size*

*Fig. 3—Below—Hardness-temperature relations of bearing alloys suitable for heavy-duty service*



Fatigue Failures: In heavy-duty bearings fatigue fa ures are produced usually by a combination of cyclic peak loa and improper or loose fits of the shells in their housings. T means that a definite bearing area is pounded by the peak lo every cycle and, if the physical properties of the bearing ma rial such as compressive strength and brittleness are inadequa fatigue failure may follow. Numerous small cracks or mosa pattern on the bearing surface are the first indication of fatig distress, *Fig.* 1. This bearing operated under high cyclic loa at lower than rated speeds. With time, and under addition pounding, the cracks will increase in number and size. Dep of these cracks also will increase and may penetrate the enti bearing-material thickness, thus reaching the bearing shell. such cases, some sections of the bearing material may be lift by the high hydrostatic oil pressure, resulting in a partial loss the bearing surface, as shown in *Figs.* 1 and 2.

### Hardness—Temperature Relation Is Important

Hardness of Bearing Materials: Chemical specific tions of several heavy-duty diesel-engine bearing materials a given in Table I. The brinell hardness of cadmium-base, co per-lead, lead-base, and the tin-base bearing materials is plott against temperature, *Fig.* 3. Since the hardness is proportion to the compressive strength, this graph is doubly important. T region of temperature of particular interest is between 200 a 300 degrees Fahr., indicated by the double lines. Cadium-ba alloys are rather hard at room temperature and rapidly lo their physical properties with increasing temperature, as ind cated by the steep curve. Copper-lead mixtures retain th hardness and physical properties almost unchanged in the r gion of temperatures plotted.

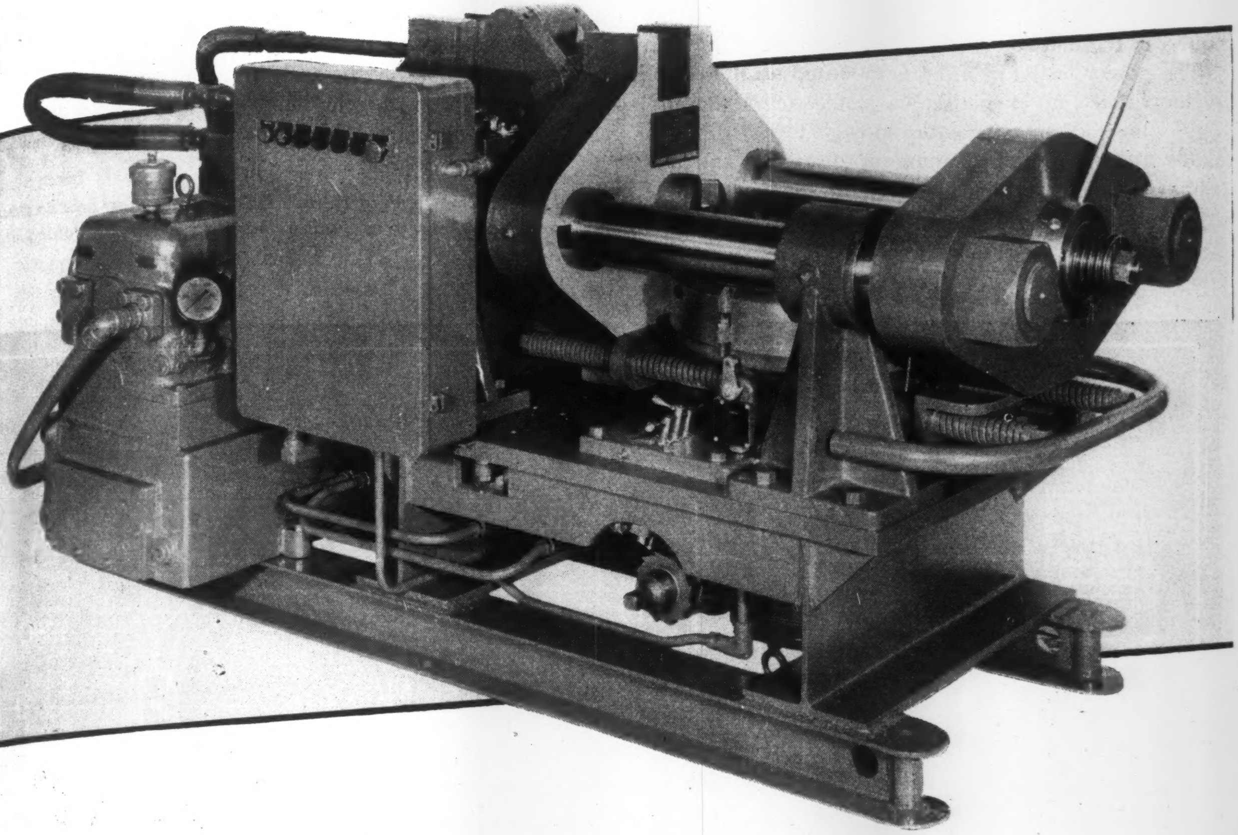
Too hard a bearing material may score the journal surfa while too soft a material may yield under the bearing pressu

### Local Hot Spots Contribute to Corrosion

Corrosion: Corrosion of bearing materials is due to che ical action of oxidized lubricating oils and can usually be ide tified by rough and pitted surfaces, such as those shown in Fi 4. In most cases corrosion occurs over small areas of the bea ing and is associated with high localized temperatures. Corr sion of Satco shown in *Fig.* 4 occurred over approximately per cent of the bearing area during the first 24 hours of oper tion of the bearing. It is believed that this was produced by small amount of sulphuric acid which formed in the crankca when sulphur trioxide from the fuel mixed with water co densate while the engine was stopped. The small surface pi shown in the photomicrograph, *Fig.* 4, are approximately .00 inch deep.

Corrosion of a cadmium-nickel bearing was recorded du ing a pretrial inspection. The most intense corrosion occurre next to the copper-deposited flange, and it is believed that co per might have acted as a catalyst during the initial running- period. This bearing was reinstalled and operated in the engi for a total of 2258 hours without any additional corrosion. It

(*Concluded on Page* 176)



# Balanced Linkage Obviates Thrust on Frame

**By Paul S. Jackson**

*President*
*Jackson Hydraulic Machine Co. Inc.*
*Rockford, Ill.*

IMPORTANCE of firmly anchoring the copper "rotating band" on large caliber Navy shells to insure that the band does not slip on the shell necessitated the development of special shell-notching machines. To produce indentations on a 90-degree angle, .05-inch deep around the shell body for this anchoring, the shell notcher in Fig. 1 was developed by the Jackson Hydraulic Machine Co. Notches are spaced approximately ¼-inch apart and, on a 16-inch shell, there are 192 grooves, each 5 inches long.

Automatic cycling is utilized in this machine which indexes for each notching stroke and stops after the entire periphery is notched. With each successive loading the operator initiates the cycle by pushbutton control. Designed for quick changeover between different-size shells by utilizing interchangeable adapter rings on the turntable and by resetting the tool holders, the machine is capable of notching shells in 4, 4½ or 5 minutes for 12, 14 or 16-inch shells, respectively.

Tool plates on the machine are hydraulically operated through a toggle-link mechanism. Design of this linkage is particularly interesting because of its balanced action on both tool members. Referring to *Figs.* 2 and 3, the power stroke of the hydraulic piston actuates a pair of linkages, at top and bottom between two movable plates, spreading them apart. One plate holds a notching tool and the other is connected by two shafts to a companion plate with another notching tool located diametrically opposite the

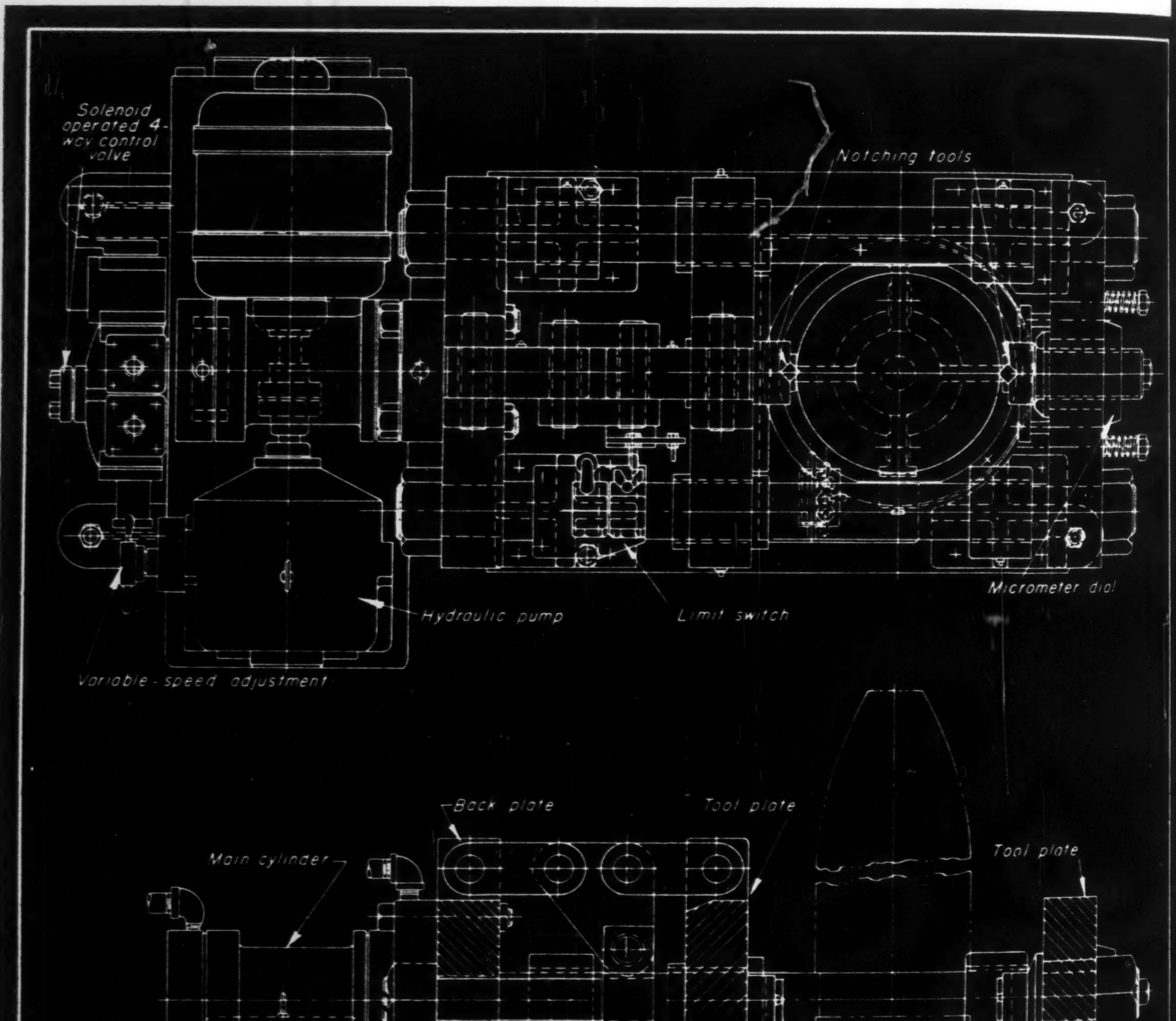
*Fig. 1—Above—Semiautomatic shell-notching machine employs double-acting toggle linkage*

first tool with respect to the shell turntable. These shafts are chrome-nickel steel bars and serve as ways for the first tool plate. They rigidly bolt the back plate to the second tool plate and are slidably mounted in bearing brackets bolted to the machine bed, *Fig.* 2.

The power-piston cylinder is mounted on the back plate as shown in *Fig.* 2 so that its piston action through the toggle links directs its entire power toward forcing the plates apart without straining or distorting the machine frame. This movement necessitates use of flexible hydraulic connections, *Fig.* 4. Because the toggle linkage

approaches a straight line during the power stroke, m
chanical advantage increases as the notching action p
gresses. When in the closed position at the end of
stroke, maximum force is exerted on the tools proportio
to the cotangent of the angle between the links. The
erating piston for the system has an 8½-inch stroke wit
7⅛-inch bore, developing 650, 750 and 850 pounds p
square inch pressure for notching 12, 14 and 16-in

*Fig. 2—Diagram of toggle linkage showing how no stres
are transferred to bed or turntable from notching act*


Solenoid operated 4-way control valve
Notching tools
Micrometer dial
Hydraulic pump
Limit switch
Variable-speed adjustment
Back plate
Tool plate
Tool plate
Main cylinder
Spring
Oil tank in base
Index cylinder
Limit switch

lls, respectively. Due to the tre- ndous pressures developed, the ee main plates are 5-inch armor te, flame cut.

With a shell in the turntable, action the linkage and the two tool plates equalized by the resistance of the ll to the notching tools. To pro- le balanced action of the tools be- e engaging the shell and after the tching is completed, adjustable rings are connected between the ol plates and the turntable. This moves any thrust on the table that ght otherwise occur from unequal tion at either end of the stroke re- lting from friction in the ways, and o assures that both tools clear the ll during indexing and loading op- ations.

Adjustment for depth of notch is ovided by a micrometer-dial ad- sting nut shown on the tool plate at e right in *Fig.* 1. This unit ad- nces or retracts its tool holder as quired for depth of indentation. Be- use of the balanced design, this one ljustment controls indentations made both tools.

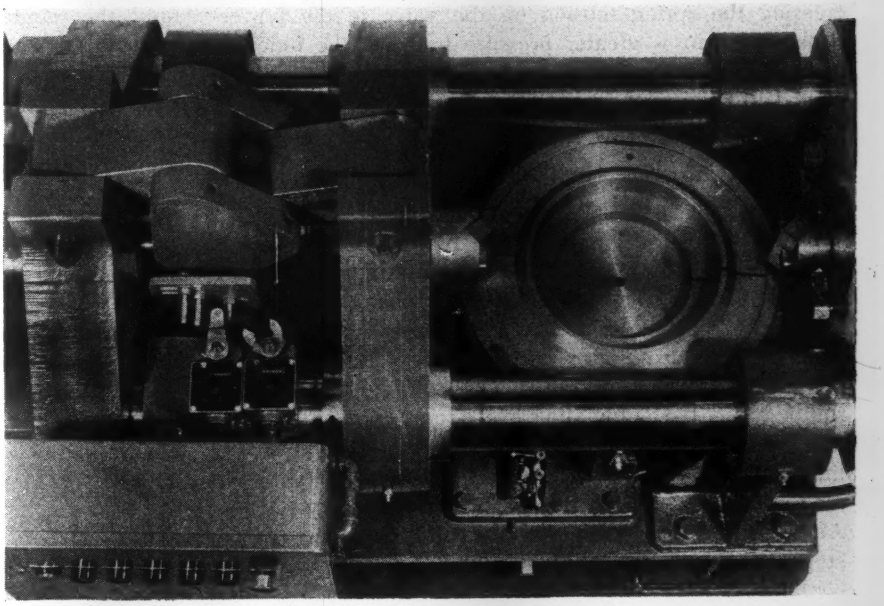
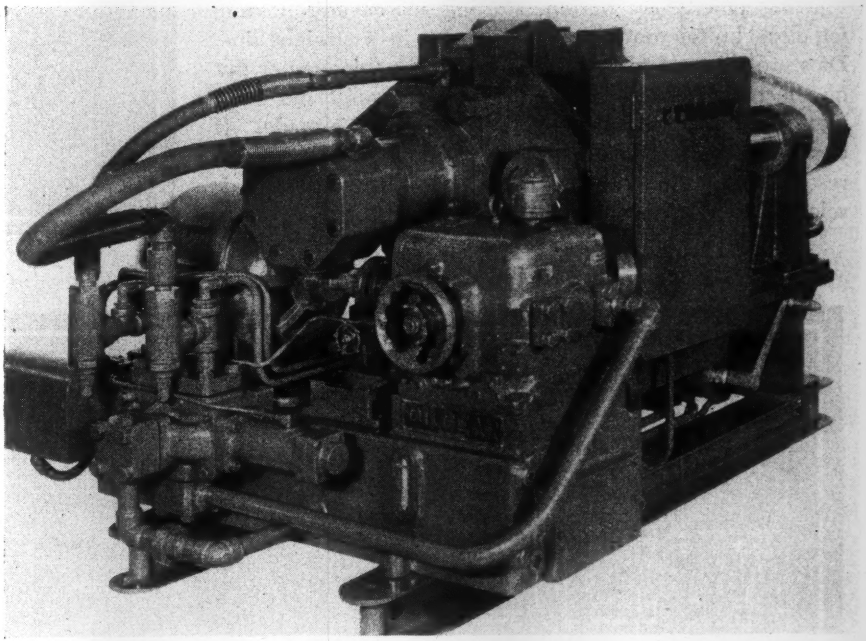


*Fig. 3—Above—Top view of turntable showing limit switches, toggle linkage and tool holders*

*Fig. 4—Below—Motor, pump, valves, main cylinder and control are mounted compactly at end of machine*



### Indexing Combined with Stroke

Table indexing between each stroke combined with the limit-switch cir- it for the tool-action cycle. Limit itches and their stops connected to e main piston may be seen in *Fig.* Indexing is performed by the ac- on of a 2-inch cylinder with a 3- ch stroke against a pawl and ratchet ared to the turntable. Selection of ratchet for each shell size is made turning one of three selector levers engage the proper ratchet. These tchets and a pawl may be seen in e bed of the machine beneath the ntable, *Fig.* 1. The selector levers e directly above, mounted on the d. To the right of the levers is a it switch for stopping the machine ter completion of the cycle. A stop is on each side of e turntable, because with the double notching action ed, the table turns only 180 degrees for a complete op- ation. To preclude possible overtravel of the turntable ring indexing, a brake is utilized in the table assembly. is a friction disk, spring loaded against the table shaft. load is just sufficient to resist rotation of the table by nd. Manual indexing for set-up work may be accom- shed by a hand crank attached to the ratchet shaft as wn in *Fig.* 4.

In *Fig.* 3 is shown a 10-horsepower, 1200 revolutions r minute motor drive connected to a variable-volume hydraulic pump. Valves and main power cylinder are also shown. The hydraulic circuit diagram is illustrated in *Fig.* 5. Adjustment for oil volume is made at the pump. It normally operates, however, at full volume, delivering 22.9 gallons per minute. Forward and return actions of the main piston and the indexing piston are controlled by a solenoid-operated 4-way valve. While cycling, this valve is reversed for each stroke by limit switch 2. When the stop on the turntable trips limit switch 1, its circuit overrides the oscillating circuit, retracting the main piston until it trips limit switch 3, stopping the machine.

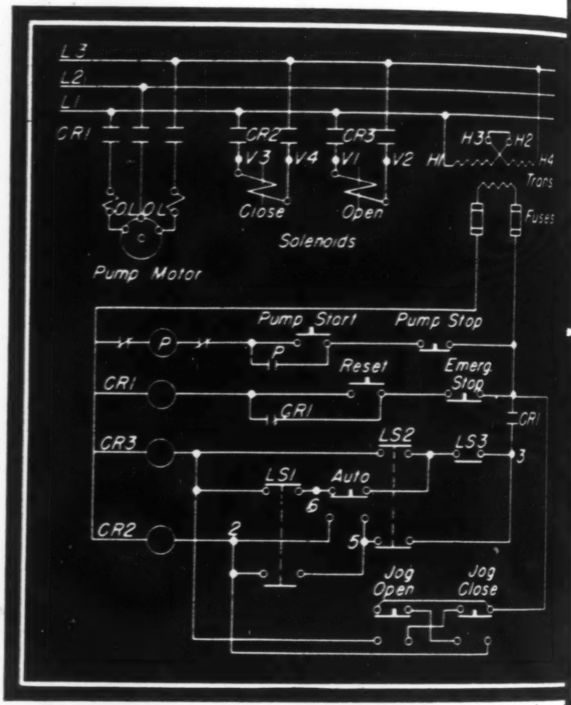
The free-flow and check valves are spring loaded. Increasing the spring tension on the valve in the power-stroke line causes greater pressure to go to the indexing cylinder. A similar valve on the return stroke line is employed to balance the hydraulic circuit.

### Operations Are Pushbutton Controlled

Electrical control circuit for the hydraulic operations is schematically indicated in *Fig.* 6. Pushbutton controls are located in the cover of the control cabinet shown at the left in *Fig.* 1. These functions as shown in the diagram include pump start, pump stop, jog open, jog close, automatic cycling, reset and emergency stop. With the pump running and the automatic-cycle circuit closed, the main and indexing pistons are cycled forward and backward by limit switch 2. A double-throw switch, it is tripped by the motion of the main piston to provide the oscillating action. After 180-degree rotation of the turntable, one of two stops on the table trips limit switch 1. This closes the contacts of *CR* 3 and the main piston retracts, opening the machine beyond its oscillating limit by overriding limit switch 2. In the fully retracted position, limit switch 3 is tripped to stop the machine giving maximum clearance for unloading and reloading the part. Operation is initiated again with the pushbutton control.

Pushbutton operation to set up the machine involves: Pressing emergency button, starting motor and holding jog-close button until toggle links are in a straight line. Dies are then set to gage distance by the micrometer nut on the tool plate. With a shell in place, reset and automatic buttons are depressed until one notch is made. If depth is correct, ratchet feed is engaged and automatic and reset buttons are again depressed until limit switch 1 is released. The machine will then index automatically until the table turns through 180 degrees and trips limit switch, completing the cycle.

With a bed fabricated of standard sections and plate, simplicity of the machine permitted it to be put into p duction quickly to meet the urgent need for shell notch Patents on the design are applied for and the company has designed a higher production machine that will dent all notches simultaneously around the shell periph and automatically apply the rotating band.


L3
L2
L1
CR1
CR2
CR3
V3
V4
V1
V2
H1
H3
H2
H4
Trans
Fuses
OL
OL
Close
Open
Solenoids
Pump Motor
Pump Start
Pump Stop
P
P
Reset
Emerg. Stop
CR1
CR1
CR3
LS2
LS3
3
LS1
Auto
6
CR2
2
5
Jog Open
Jog Close

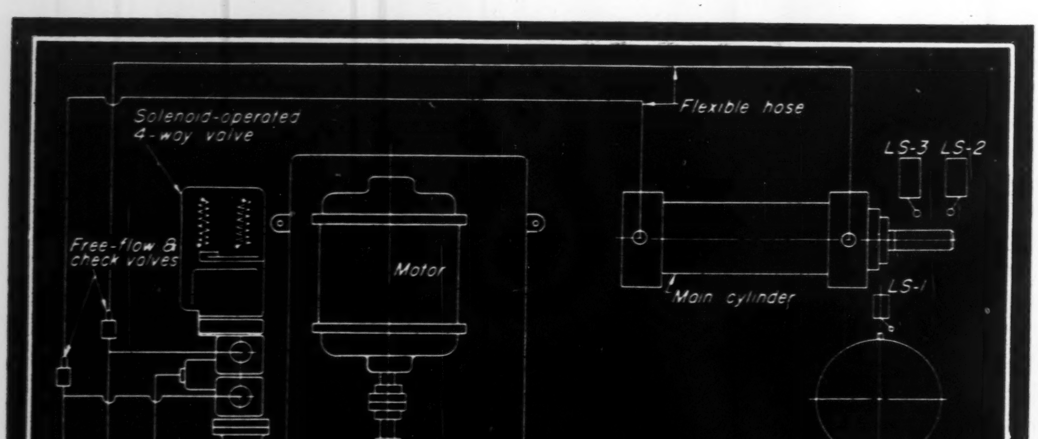
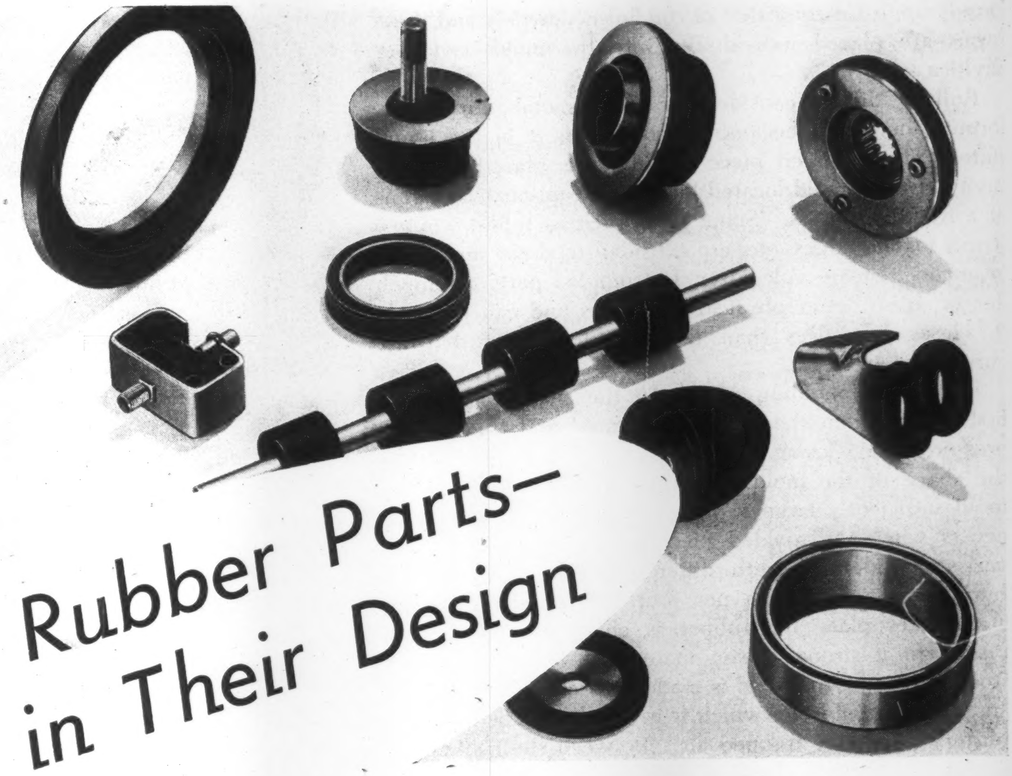

Solenoid-operated 4-way valve
Flexible hose
LS-3
LS-2
Free-flow & check valves
Motor
Main cylinder
LS-1
Index Table
Index cylinder
Pump
Index ratchet
Hand wheel for oil volume control
B
A

*Fig. 5 — Left — H draulic circuit notching and indexi Limit switches cont position of 4-way val Circuit is balanced free-flow and chec valve units*

*Fig. 6—Above—Sch matic of control circ for operation of pu motor and the 4-w solenoid valve*

# Molded Rubber Parts— Factors in Their Design



, 1—Representative
up of rubber-to-
tal bonded machine
parts

**By Curt Saurer** and **J. F. McWhorter**
*Consulting Engineer* *Research Engineer*
*The Ohio Rubber Company, Willoughby, Ohio*

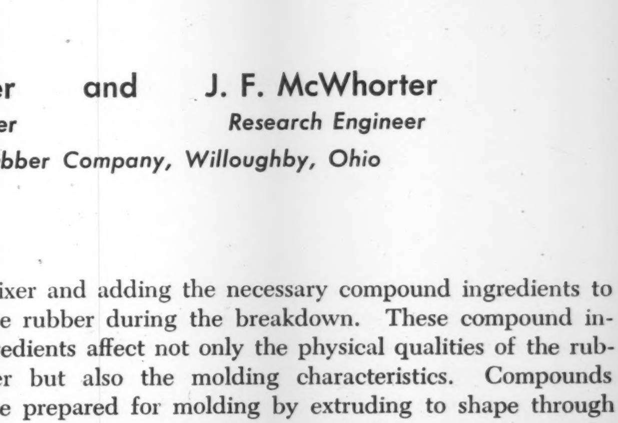
ABILITY to absorb energy makes rubber an outstanding material for insulating and damping vibration and noise. Although machine designers have een aware of this property of rubber ever since the dis- very of vulcanization, it has been only during the last o decades that they have used rubber extensively in e design of vibration insulators, cushioning drive mem- rs and noise eliminators.

Probably this has been due to the fact that parts de- gned for these purposes generally are rubber-to-metal nded and good adhesion of rubber to metal was not hieved until about twenty years ago by a special method brass plating the metal. Modern techniques in molding d bonding, together with improvements in rubber it- lf, now make possible many designs that years ago ould have been considered impracticable. Some typical olded and bonded rubber-to-metal parts are shown in ig. 1. It is the purpose of this article to discuss the portant molding, bonding and quality factors as they fluence the actual design of the molded parts.

THE MOLDING PROCESS: As might be expected, the proc- ss by which rubber parts are molded exerts significant fluences on the actual design of the parts. Thus, be- re considering design principles, a brief discussion of e molding process seems in order.

Crude and synthetic rubber compounds are prepared breaking down the rubber on a mill or in a Banbury mixer and adding the necessary compound ingredients to the rubber during the breakdown. These compound ingredients affect not only the physical qualities of the rubber but also the molding characteristics. Compounds are prepared for molding by extruding to shape through a die, sheeting on a calender and clicking, or die punching. The prepared raw stock is similar to the finished article in density and has the consistency of a semi-hard tough, putty-like substance. It is worked into forms

*Fig. 2—Below—Sharp corners of unit at a would cause "feathering". Design at b with edge radii is preferable*



closely approximating that of the finished article and these forms are placed individually into the mold cavity or cavities.

Rubber differs considerably from metal during the forming operation because at no time is it in the liquid state. The prepared piece of rubber is placed into the cavity of a hot mold located between steam-heated platens of a hydraulic press. Some parts such as tubes, window strips, sealing edges, etc, are extruded to shape and cured in an open steam vulcanizer, but molded parts are much denser, have better physical properties and can be made to closer tolerances than the extruded parts that are cured in open steam.

During the molding operation, the rubber compound first softens from the heat of the mold and then, as the pressure upon closing the mold is applied, it is formed to the shape of the mold. Care is exercised to make the mold sufficiently large to take care of the shrinkage that occurs when the molded piece is cooling. Contraction varies considerably with different types of compounds but averages 3/16-inch per foot. As the curing operation takes place the rubber is changed from a weak plastic to a strong elastic material. To permit excess stock to escape, the mold is made with an overflow cavity, an additional effect of which is to eliminate flow lines and blisters caused by trapped air. Between the rubber part and the overflow cavity is a thin flash that must be trimmed off after the part has cooled.

Molding of Rubber and Plastics: While there are essential differences in molding rubber, plastics and metals, the process of molding rubber is more closely allied to that of plastics than of metals.

In molding rubber and plastics parts, a steel mold invariably is used, heated to from 250 to 350 degrees Fahr., the required temperature range for rubber or plastics curing. In the case of molds for plastics, the cavity is filled with preformed powder or granular material. In both cases the heated molds are closed under pressure, the overflow of stock pushes the trapped air from the cavity, and the excess stock flows into the overflow groove provided around the entire periphery of the cavity.

### How Rubber and Plastics Molding Differ

Chief differences in the molding of rubber and plastics parts are: Molds for plastics usually are made of hardened steel, highly polished. Molds for rubber are made of soft steel which, though polished originally, cannot maintain polished surfaces over a long production range. Pressure on molded rubber parts usually runs between 500 to 1000 pounds per square inch, while on plastics its usual range is from 1000 to 4000 pounds per square inch. The pressure is maintained during the entire process of curing and, generally, no cooling of the molds is necessary for rubber and thermo*setting* plastics, although molds do have to be cooled for thermo*plastic* materials before the piece is removed. The curing temperature range for rubber and plastics molding is practically the same.

It might be well at this juncture to point out that large quantity production injection-molding methods for rubber and plastics may replace the present practice of compression molding. Injection molding is similar to pressure die casting with the exception that heat is applied to the molds for rubber and plastics, while in die casting on cooling exists. The cycle in injection molding of rubb and plastics is still considerably longer than the cycle die casting but much shorter than compression molding rubber and plastics. Injection molding of thermo*plast* plastics—an old art—is relatively simple, the heated m terial being pushed into a chilled mold which sets t piece immediately. However, injection molding of rubb and thermo*setting* plastics requires that the material preheated to a temperature just under that needed f curing. This preheating allows the material to go throug a quick cure after injectiion into the mold.

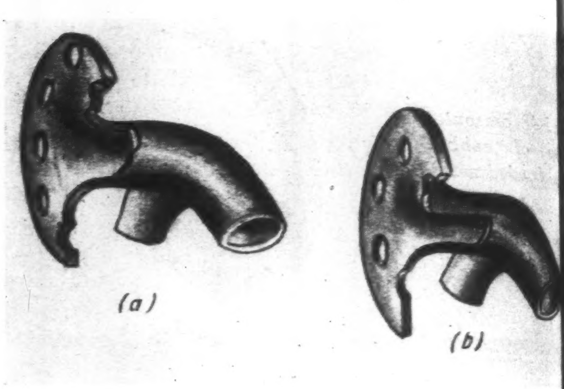
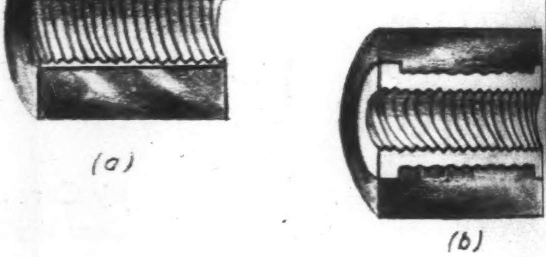


*Fig. 3—Above—Holes in part at a are too close toget and too close to edge. Better spacing is shown at*

*Fig. 4—Below—Molded rubber parts with metal inser In design at a, rubber may flow into threads during moldi Flanged insert of design b obviates this difficulty*

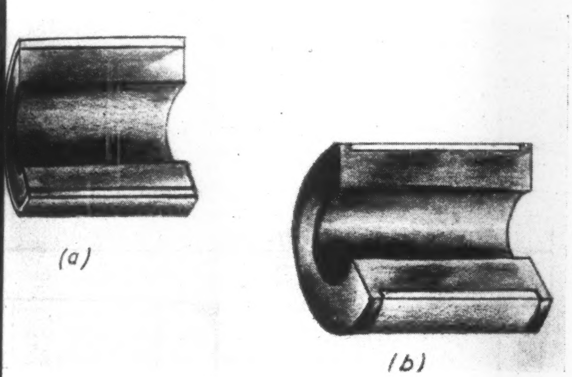


Design Considerations: The final design of a molde rubber part in many cases is a compromise between wh the design engineer wants and what the rubber manu facturer can give him economically. It also depend largely on the production quantities required and th method employed for arriving at the lowest productio cost. If only a few pieces are required of an item, single-cavity mold is made. This is comparable to foun dry molding or casting of one part. If, however, larg quantities are required, as many as 360 cavities may b made in one mold.

Molded rubber parts require a design technique tha varies somewhat from that utilized in the design of meta or plastics parts. For example, in *Fig. 2a* the sharp co ners shown would be a constant source of trouble due t "feathering" of the edges and trapped air. Unless requir ments absolutely dictate the use of sharp edges on a pro it is advisable to specify radii. In *Fig. 2b* is shown a cided improvement over the design in *Fig. 2a*. When dercuts are necessary, the designer should be certain t the article can be removed from the mold or mandrel. Holes for attaching the molded parts to other machine mbers should be kept as far from the edges as possible avoid tearing. In addition, the holes should be spaced nerously from each other. In the part pictured in *Fig.* the holes are too close to the edge and too close to- her. *Fig. 3b* illustrates good practice in hole spacing.

A typical part having a molded-in threaded metal insert shown in *Fig. 4a*. In this design the rubber under pres- re would flow around the insert and into the threads, the sult being that the threads would have to be cleaned ter vulcanization. A better way of designing this part is own in *Fig. 4b* where a heavier insert with end flanges employed. The larger these flanges can be made, the s the rubber that will flow into the threaded hole. To improve adhesion of the rubber to the metal, it is well to have a rough or corrugated surface on the outside diameter of the insert.

*g. 5—Below—Rubber parts with metal shells molded in place. Design b is superior to that shown at a*



### Size and Strength of Shells Important

In *Fig. 5a* is shown a molded part with a "molded-in" metal outside shell. Designers generally draw up such parts with, for example, a plus or minus .005-inch tolerance in length and diameter. From the rubber manufacturer's standpoint, however, a tolerance of plus zero minus .01-inch on length and diameter is more desirable. Also, the shell will have to be made strong enough so that the pressure of the rubber will not stretch the shell to a larger diameter and prevent its removal from the mold. A preferred design is shown in *Fig. 5b* where the shell is slightly shorter, giving the rubber a better chance to hold.

Overflow flash exists on all molded parts and, as has been indicated, is trimmed off after the parts have cooled. From a production standpoint it also is important to have this flash at a point where it is easiest to remove either by machine or by hand trimming with shears.

Illustrated in *Fig. 6a* is a design in which the rounded corners require that the flash be located at a point where it is difficult to cut with a knife or trimming die without injuring the finish or contour of the piece. In a case of this type, buffing the flash off would be the accepted practice. A buffing operation, however, is not as fast as knife, die or shear trimming and therefore is more costly. In many cases it is possible to remove flash by a tumbling operation but this, of necessity, spoils the finish on a part. *Fig. 6b* shows a similar part where the flash is at an outside edge, making it easily removable.

Mold design varies greatly with the part required—from

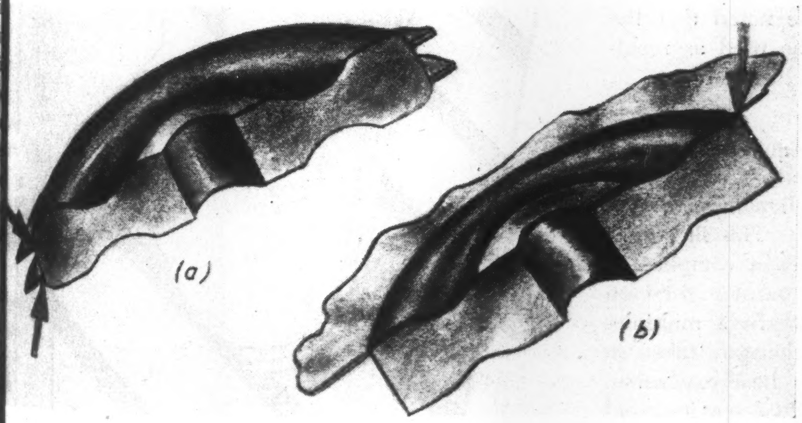
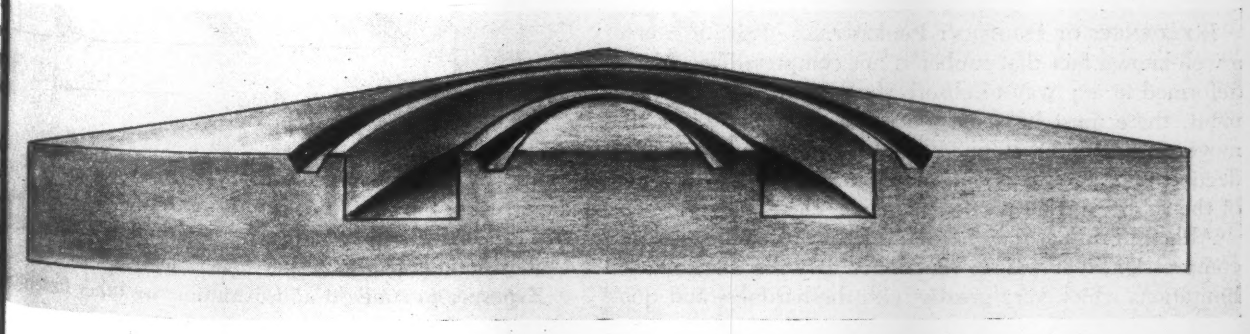


*Fig. 6—Left—Location of flash is important. Flash located as shown at a is difficult to trim from part. Flash at outside edge of part, as at b, is more easily removed*

*Fig. 7—Below—Mold for producing simple molded ring has cavity and overflow grooves cut on one side*



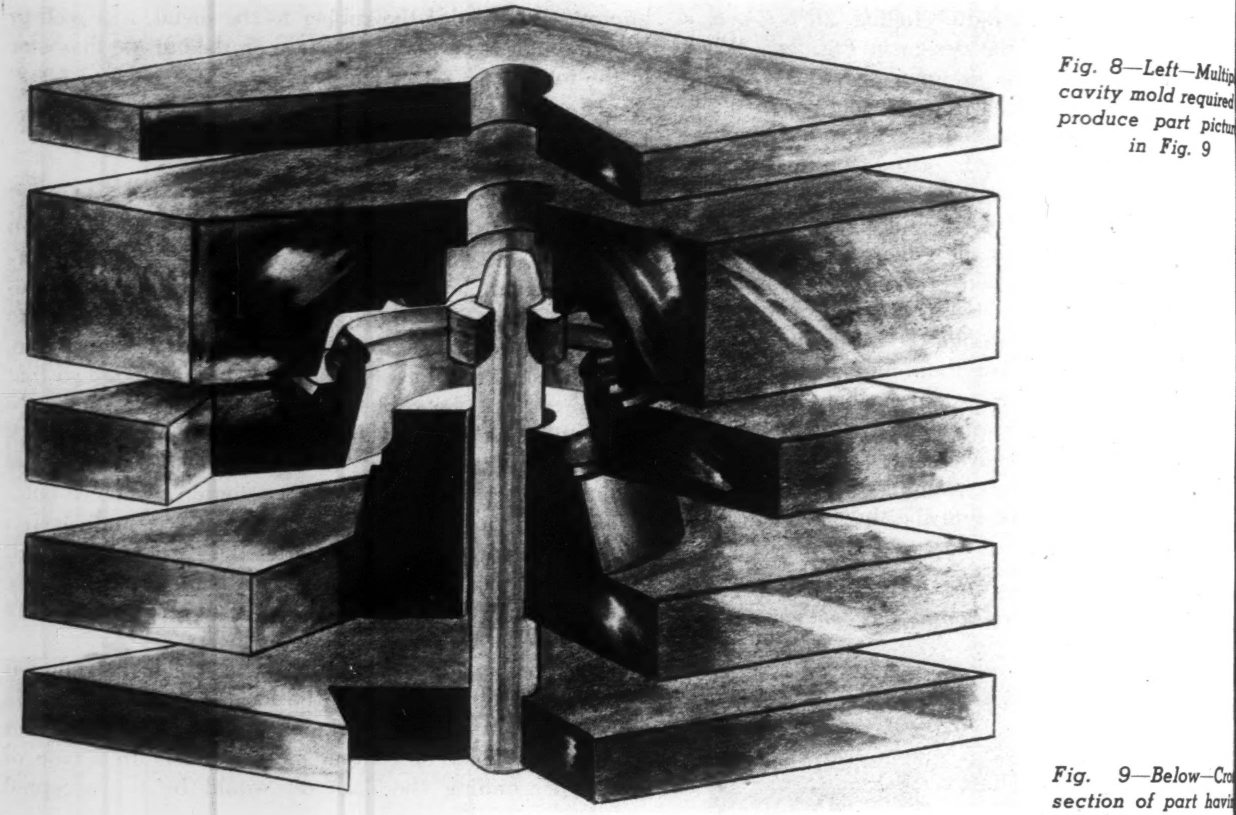


Fig. 8—Left—Multip
cavity mold required
produce part pictu
in Fig. 9

Fig. 9—Below—Cro
section of part havi
molded-in metal inse

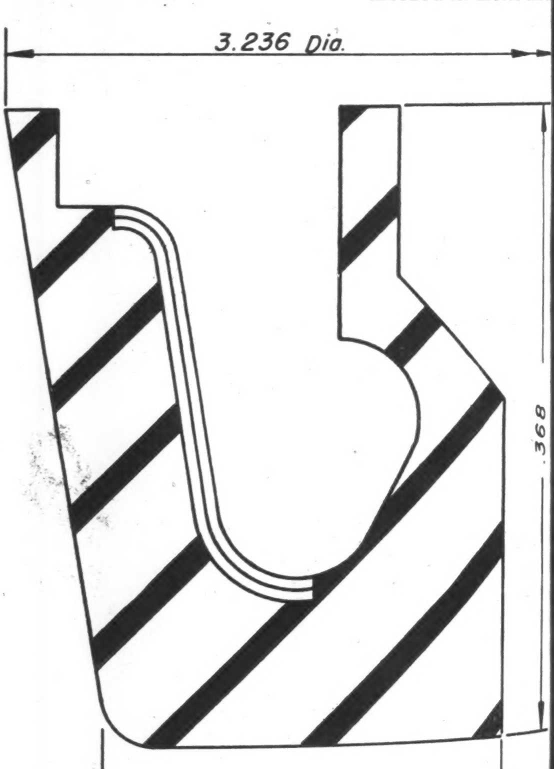
a one-plate mold to five or more plates with necessary removable mandrels for undercuts or holes. Naturally, the simpler the part the more easily it can be produced and the lower the cost. *Fig.* 7 shows a mold for producing a simple ring. In this case the cavity and overflow grooves all are cut on one side of a steel plate, which constitutes the mold, and a loose top plate is placed on the mold to serve as a closing platen. It should be noted that the platens of the hydraulc press cannnot be used as mold-closing plates.

### Complex Molds Sometimes Required

To produce the part pictured cross-sectionally in *Fig.* 9 requires the complicated mold of *Fig.* 8. This illustrates how a relatively simple part necessitates a complicated mold wherein the different mold platens must register on top of each other with great accuracy. In a multiple-cavity mold of this type, special precautions are taken so that the cavities will not vary due to heat expansion. Where accuracy of the part from a multiple-cavity mold is required, the cavities generally are registered individually with each other.

Influences of Inherent Properties: It is, of course, a well-known fact that rubber is not compressible. If it is deformed in any way to absorb shock or to take up movement, there must be some place for it to go. Thus, if a movement is to be stopped at a definite point in the deflection of a part, it is only necessary to restrain the flow of the rubber at that particular point.

Although rubber may be deformed in tension, shear and compression, it is well to remember that there are natural limitations which vary greatly with the hardness and quality of the compound used. Temperature changes ca have severe effects if the rubber or rubber-like material not especially compounded to be practically immune heat variations within a specified range.



Exposure to sunlight and oxidation are other factors th ıst be taken into consideration and may require special mpounding. Then there is frequency of flexing or im-ct, and the amount of the deflection or intensity of im-ct. Still further, the part must be designed so that it ll not overheat and destroy itself. Parts that tend to at up are those which are called upon to withstand con-lerable, repeated deflection. In this type of application is well to reduce the amount of deflection by increasing e amount of rubber.

It is essential that the method of fastening rubber parts other members be such that no chafing can occur at the eas of contact. Parts of this type should be designed so at all distortion or flexing occurs in the main body of the bber itself and not on the attaching surfaces. A good le to follow for any given movement or deflection is to e a column of rubber four times the extent of the move-nt. For example, if a rubber device must deflect 1/4-ch, a column of rubber not less than 1-inch gh should be used.

Parts designed with due consideration ac-rded the various factors discussed in the regoing will last indefinitely. In fact many bber parts have been performing satis-ctorily without undue permanent set for 20 ars or longer. Machine designers will find wise to describe on their drawings the action and conditions under which the rubber part s to function, thus enabling the rubber engi-ers and chemists to prescribe the most suit-le compounds for specific jobs.

FASTENING MOLDED RUBBER PARTS IN PLACE: or years, where rubber parts have been used all kinds of machines, the fastening means mployed generally has been a metal flange ith a rounded bead to hold the rubber against e machine part surface (shown in *Fig.* 10*a*). hen these rubber parts had to do some actual work a afing occurred on the surface and fabric reinforcements ere therefore used on the attaching surfaces to impart nger life.

## Advantages Offered by Molded-in Fasteners

With the introduction of rubber-to-metal adhesion, uch more satisfactory parts could be made at lower as-mbly costs by vulcanizing the rubber directly to a metal stening plate. Holes or integrally fastened bolts (*Fig.* 0*b*) are optional. This undoubtedly is the preferred ethod today and can be used in a variety of applications here the bond between the rubber and metal is stronger an the rubber or synthetic rubber employed. When it necessary to fasten metal parts together with rubber provide a flexible connection or coupling, it is possible combine compensation for several movements in one it. This also offers the advantage of incorporating flexi-lity in one direction and relative rigidity in another di-ction in a single unit.

INFLUENCES OF COMPOUNDING: Generally, to obtain e greatest cushioning effect and maximum life under namic movement, a soft rubber compound is best and large volume is better than a smaller amount of harder mpound. With a soft compound, however, load-carrying capacity, physical properties and aging time are reduced and manufacturing difficulties increase.

It may be expected that the cushioning qualities of synthetic rubber, especially under dynamic loading conditions, will be inferior to some extent in most applications to those of natural rubber. However, this situation undoubtedly will be improved as experience is obtained in design and as improvements are effected in the manufacture and compounding of the synthetic polymers.

## Synthetic has Superior Chemical Properties

Offsetting its limitations in physical properties, synthetic rubber offers advantages over the natural products in chemical properties. For example, the use of synthetic rubber in place of natural has made it possible to design machine parts that will not decompose in oils and greases and will successfully withstand deterioration resulting from natural aging.

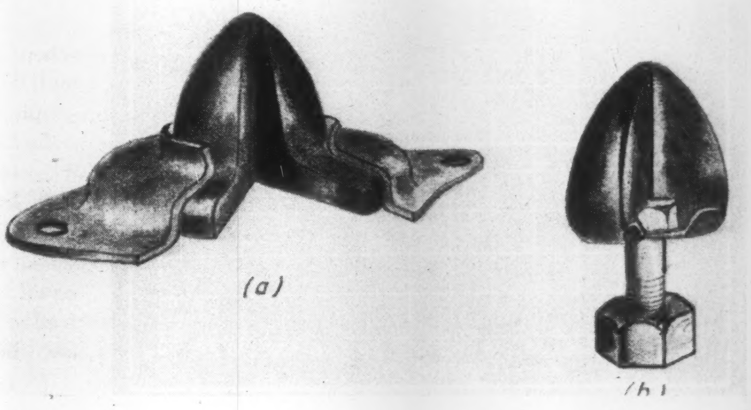


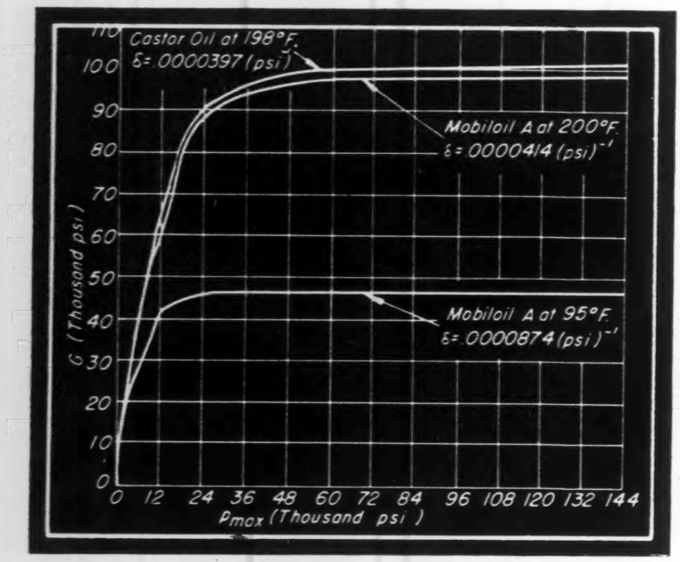
*Fig. 10—Holding rubber parts in place with beaded metal flange, as at a, is being superseded by molded-in fastening means as at b*

While synthetic compounds can be produced in the same hardness as natural rubber compounds, their tensile strength, permanent set and elongation may be reduced so severely that they are impractical for general use. Thus changes in design become necessary to achieve the same results. To attain the same dynamic deflection, for instance, harder synthetic compounds have to be used, requiring reduced cross-sectional areas with consequent increase of internal stressing. Under dynamic conditions, generally speaking, the flex life is much less and the heat generated much greater in synthetic compounds than in natural rubber compounds of the same hardness. Such changes may be noted in automotive mountings and bumpers.

The rubber industry has made tremendous progress with synthetic rubber compounds in the last three years and there is just cause for confidence that further chemical and technical developments will bring us eventually to the point where synthetic compounds can duplicate the performance and strength of soft natural rubber compounds. The machine designer is well advised to maintain close contact with his supplier and take advantage of all the new developments and improvements that are constantly taking place in the rubber industry and which hold such rich promise for the future.

# Load Capacity of Gear Tooth Oil Film

### By E. K. Gatcombe



EXPERIMENTAL work on gears has indicated the desirability of establishing and maintaining an oil wedge between gear-tooth surfaces for the mating period. Through the use of hydrodynamic theory—subject of course to certain assumptions—it can be shown that an oil wedge of a reasonable minimum thickness may be established for each mating phase. The analysis* yields the following final equation for the load-carrying capacity of the wedge per unit face width:

$$F = k_0 L_c \cos\alpha \, \frac{\pi}{24}\left(\frac{81\psi}{20} + \frac{\pi G \log_e 10\sqrt{2rh_0}}{15}\right) \quad \ldots\ldots\ldots (1)$$

where $F$ = Pitch line component of the total supporting action of the oil wedge

$k_0$ = Ratio of average supporting action of the wedge to that at the pitch-point phase, which may be taken as 1.3

$L_c$ = Average number of pairs of teeth making simultaneous mesh

$\alpha$ = Pressure angle

$$\psi = \frac{4}{3}\sqrt{2rh_0 \; p_{max}}$$

$$r = \frac{R_1 R_2 \sin\alpha}{R_1 + R_2}$$

where $R_1$ and $R_2$ = Pitch radii of pinion and gear, in

$h_0$ = Minimum oil-wedge thickness, in

$p_{max}$ = Maximum pressure in film, psi

$G$ = Factor depending on $p_{max}$ and characteristics.

Procedure in checking the load-carrying capacity o pair of gears, from the viewpoint of lubrication alone as follows:

1. Select an oil of known viscosity
2. Compute $L_c$ from the geometry of the gears
3. Choose a value of the maximum permissible surf pressure, $p_{max}$, and find the corresponding value of from graphs similar to the accompanying chart
4. Compute the value of $h_0$ from

$$h_0 = \sqrt[3]{\frac{2r(12\mu_a U)^2}{G^2}}$$

where $\mu_a$ is the viscosity of the oil at atmospheric press and the stated temperature, and $U = \omega_1 R_1 \sin\alpha$ where = pinion speed, radians per second.

5. Compute the value of $\psi$ from the relationship pre ously given
6. Solve Equation 1 for the load-carrying capacity.

It should be noted that the wedge may break do when its minimum thickness, $h_0$, becomes approximat equal to the surface roughness. Maximum pressure in wedge may approximate the value of the surface end ance limit of some gear materials.

EXAMPLE: A 3140 SAE steel straight spur pini (bhn = 325), 2-inch pitch diameter and 1.625-inch wid of face meshes with a 1045 SAE steel gear (bhn = 250 10-inch diameter. Teeth are 20-degree involute st diametral pitch 12. It is desired to calculate the oil-wed carrying capacity at 1750 revolutions per minute pini speed. Solution is as follows:

**1. Use Mobiloil A for which $\mu_a = 5.5\times10^{-6}$ pou seconds-inches$^{-2}$ at 95 degrees Fahr. and $\delta = .00008$ (psi$^{-1}$). $\delta$ is slope of the pressure versus log viscosity cur**

**2. From geometry of gears, $L_c = 1.23$ inches.**

**3. Maximum permissible surface compressive stress 56,000 psi will be selected for $p_{max}$. Corresponding val of $G$ (see chart) is 46,000 psi.**

**4. $U = 1\times.342\times183.3 = 62.6$ inches per second**

$$r = \frac{1\times5\times.342}{1+5} = .29\text{-inch.}$$

$$h_0 = \sqrt[3]{\frac{2\times.29(12\times5.5\times10^{-6}\times62.6)^2}{46{,}000^2}}$$

$$= .0000168\text{-inch (satisfactory)}$$

**5.** $$\psi = \frac{4}{3}\sqrt{2\times.29\times.0000168\times56{,}000}$$

$$= 232 \text{ pounds per inch}$$

**6.** $$F = 1.3\times1.23\times.94\times\frac{\pi}{24}\left(\frac{81\times232}{20} + \frac{\pi\times46{,}000\times2.3\times.0031}{15}\right)$$

$$= 197 \text{ pounds per inch}$$

For total face width the force is 197 × 1.625 = 320 pound

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*The analysis is discussed fully in the author's recent A.S.M.E. paper "Lubrication Characteristics of Involute Spur Gears", of which this article is an abstract.

# mproving Design of Paper Cutters

### By H. C. Cook*

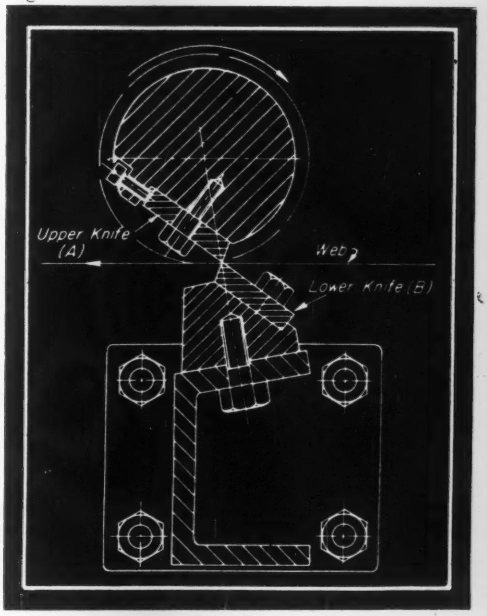

*Fig. 1—Cross section through web of material to be cut, and lower and upper knives*

T IS not uncommon for revolutionary changes to result from the reintroduction of old established principles in the application of modern scientific design. Such a rebirth is the recent introduction to the per-converting industry of the cross-cutting method applied to rotary b-feed machines whereby two knives, one stationary under the web, d the other rotating above the web, effect the transverse cutting while e web is moving in its passage through the machine, *Fig.* 1. With a oper understanding of the principles, as explained in this article, this vice may be used to advantage in many applications where it would herwise have been considered impracticable, particularly in cutting b-lengths within a range of sizes of small dimensions.

A rotating knife which has a transverse cutting edge traveling at the me speed as the web during the period of cutting will effect a per-ndicular transverse cut in the web. In this simple form, the cut length termines the diameter of the knife, which is timed to rotate at uniform eed, making one complete revolution for each cycle of the machine, shown in the illustration *Fig.* 1.

It is common practice to make one knife, eferably the lower one, of hardened steel, hile the upper, rotating knife is formed some softer material such as mild steel or onze. In setting up, it is only necessary bring the two knives carefully into con-ct during rotation of the machine. This fects a trimming operation to the softer ife and thus insures a continuous trans-rse cut along the whole length of the op-sed edges. It is to be noted also that notching or intermittent transverse cutting may be arranged easily by cutting away portions in the severing edge of the softer knife.

An instantaneous cut is avoided by setting the two knives out of parallel, thus obtaining a shearing or scissor-like action and smoother operation of the machine. Conventionally, the lower knife only is set at a slight angle from the perpendicular to the direction of web travel, *Fig.* 2. The amount of obliquity of this knife has no detrimental effect on the squareness of the resultant cut, provided the peripheral speed of the rotating square-set upper knife is identical with that of the traveling web.

Referring to *Fig.* 2, which shows diagrammatically a plan view of the device, commencement of the cut takes place between the knife edges at *X*, but as the upper knife sweeps over the lower to complete the severance at *Y*, the web has moved the same distance as the knife, *X* to *Z*, and the commencement of the cut has progressed during cutting to arrive at a point opposite that of final severance *Y*. Thus a square cut is assured.

In the case of machines requiring one cut length, correct speed is insured simply by making the periphery of the cutting-knife path equal to the length to be cut. If the cut length is to be adjustable, a common expedient is to arrange a pair of elliptic gears in the driving train to the knife. By this means the speed of the knife edge is made to vary during each revolution between a minimum and maximum, and the relation between the elliptic gears and the cutting edge of the knife is adjusted to select the required speed within this range, so that the knife edge has a speed approximating that of the web during the period of cutting.

Certain disadvantages, however, are inherent in this arrangement. First, the adjustment of the elliptic gears presents difficulties to the average machine operator. Also the design of the knife is restricted through being determined by the cutoff lengths. Its average circumference, when multipled and divided by the elliptic gear ratio, must provide dimensions which agree with, or preferably extend slightly be-

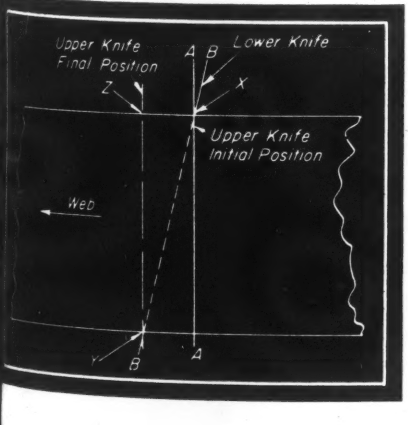

*Fig. 2—Plan view showing conventional arrangement of knives with angle of set greatly exaggerated*

yond, the sizes of the maximum and minimum web cut-off lengths required. In effect the diameter, which must be maintained as large as possible to provide rigidity, is governed and restricted in a manner which makes this arrangement impossible for small cutoff lengths. An alternative is to provide two or more knife blades on the rotating holder, which is timed to run at the appropriate reduced speed. The effective knife diameter may thus be increased by the same ratio but the maintenance of several cutting edges is not conducive to easy setting, also the construction of the knife-holder becomes complicated as each blade must be provided with both adjusting and locking screws.

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*This article is based on the author's experience Chief Designer with J. Dickinson & Co. Ltd.

Introduction of an angular setting to the upper knife blade opens up a field of possibilities for overcoming these difficulties. Considering *Fig.* 3, the components are shown as in the previous figures but with the upper knife *A-A* set at an angle to the transverse cutting line required. Here the commencement of the cut takes place between the knives at *M* and as the upper knife sweeps over the lower to complete the cut at *N* the web must travel from *M* to *N* to insure that the transverse severance of the web is square. While the web has traveled this distance, the upper knife edge has progressed from *M* to *Z*, a distance equal to the sum of the angular set of both upper and lower knives. The distances *M* to *N* and *M* to *Z* are obviously relative to the respective velocities of the web and the upper knife, and the distance *M* to *N* also represents the angular set of the lower knife. Thus:

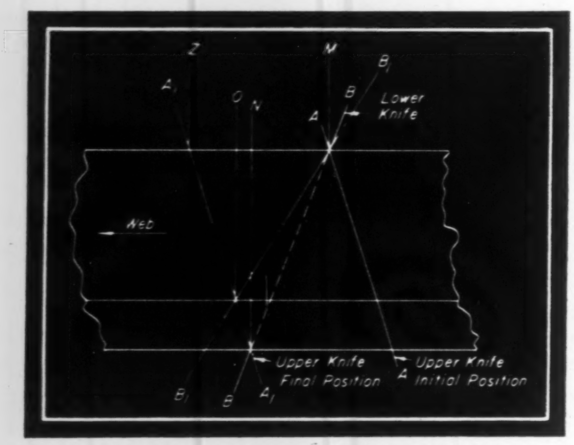

***Fig. 3—Improved arrangement using angular set for both knives facilitates cutting a range of lengths. Angles of set are exaggerated***

$$\frac{MN}{MZ}=\frac{\text{Speed of web}}{\text{Speed of upper knife edge}}$$

$$=\frac{\text{Angular set of lower knife}}{\text{Sum of the angular sets of both knives}}$$

Important consequence of this arrangement is that the speed of the upper knife, which of course is relative to its diameter, may be very high in comparison with that of the web, provided the angular dispositions of the upper and lower knives are arranged to satisfy the conditions of the foregoing equation. The ultimate limit is reached when the web speed is zero and distance *M* to *N* is zero, m ing a square-set lower knife and upper knife with angle and velocity.

Three methods of arranging the two proposed knive satisfy the foregoing conditions, with the further requ ment of cutting the web in a range of different lengths, indicated in TABLE I.

From the point of view of machine adjustment and of design, it is evident that the arrangement represe by Method 1 and illustrated in *Fig.* 3 has considerable

### TABLE I

### Methods of Varying Length of Cut

| Method | Upper Blade | | Lower |
|---|---|---|---|
| | Angle | Speed | Ang |
| 1 ........ | Fixed | Constant | Adju |
| 2 ........ | Fixed | Adjustable | Fixed |
| 3 ........ | Adjustable | Constant | Fixed |

vantages over the other two. The drive to the upper k is by ordinary concentric gears and the alteration in lower knife blade angle may be effected easily by sc adjustment.

In Method 2, although the two knife angles do not quire adjustment, the accurate setting of the elliptics give the desired results requires a certain degree of s Further, the uneven momentum of a massive upper k holder driven by elliptic gears does not help in the smo running of a fast machine. In Method 3 certain c structional difficulties are involved in requiring the dri upper knife to be mounted for angular adjustment. ternatively, the blade must be mounted for angular justment upon the holder.

To illustrate a practical example of the use of the sche shown in *Fig.* 3, it is required to ascertain knife prop tions for a machine having a range of cutoff lengths fr 1½ to 2½ inches, sizes which normally are beyond t range of rotary-knife cutting. Upper knife diameter su able for the design of the machine in question is selecte say 2 inches, also a suitable angular set for the blade the holder, say ⅛-inch. It is now necessary to calcula the range of angular movement necessary to the low knife to provide a square cut for the minimum and ma mum cut-off lengths.

Letting $x$ = angular set of lower knife, substitution the equation already established of a 1½-inch cutoff leng gives

$$\frac{1\frac{1}{2}}{2\pi}=\frac{x}{1/8+x}$$

from which $x$ = .04-inch minimum.

For 2½-inch cut-off length, similar substitution results $x$ = .065-inch maximum.

These figures demonstrate the possibilities of the kni proportions selected in this example. With an upper kni diameter of 2 inches the average cutoff length with th normal arrangement, *Fig.* 2, would be around 6 inches.

It will be evident that free choice of the knife diamet in the design of machines requiring a cutting couple this type, unrestricted by the cutoff lengths required, o fers greater freedom in design and makes possible the u of a rotary knife under circumstances where it would othe wise be impracticable.

# Working Stresses for Helical Springs

By A. M. WAHL
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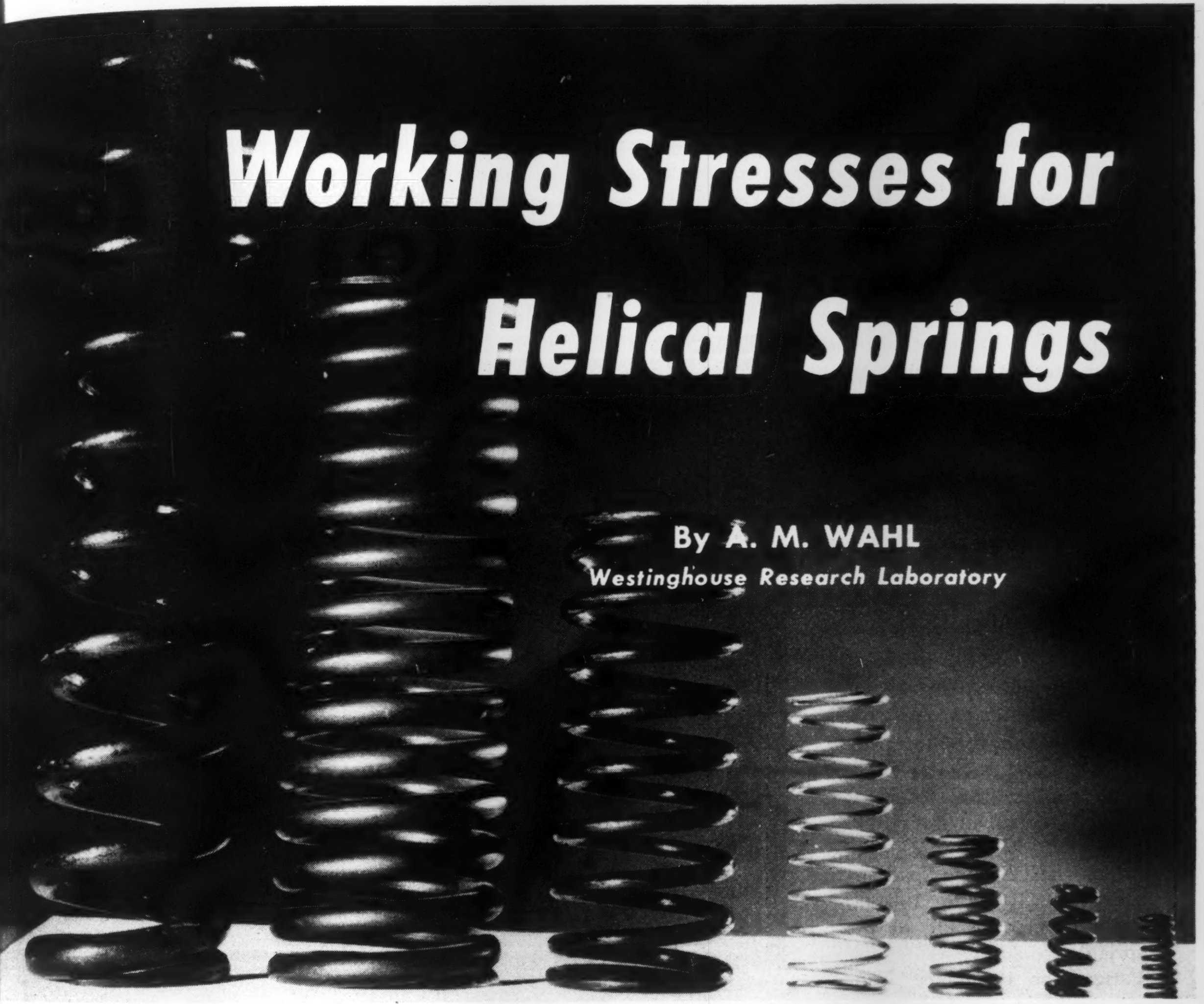

N DESIGNING a helical spring for any given application or service, the most important consideration perhaps is the choice of the working or allowable ess. This is particularly true since the size of spring reired for a given service depends in a large measure on e working stress which may be used. To meet space quirements in many practical applications it is desir- le to use as high a working stress as possible, consistent th safety. Review, however, of the literature and man- cturers' publications shows considerable variation in ggested working stress values which sometimes are con- sing to engineers.

It is the primary purpose of this article to discuss some the fundamental principles underlying the choice of rking stresses in springs in order to throw some light on e reasons for the wide variation in working stress used practical design. Particular emphasis will be given to e significance of the various stresses calculated by meth- s in common use. Utilization of these fundamental prin- ples in practical design will be illustrated by reference actual designs of springs used in industrial and railway rk. The present discussion will also be limited to steel nd-wire springs and to cases where normal temperatures and no corrosion are involved. Values of working stresses used by the Ordnance department and by spring manufacturers are given.

In helical spring design it is common practice to compute stresses in springs by one of two methods. In the first method, the spring is considered as essentially a round bar under a torsion moment equal to load times means coil radius, stresses due to direct shear and curvature being thus neglected. This method yields an average stress over the surface of the bar or wire which is often referred to as the *uncorrected stress*. In the second method, the spring is considered as essentially of helical form under an axial load with stresses due to curvature and direct shear being considered. This method, which involves elastic theory, yields the peak stress known as the *corrected stress* which is localized on the inside of the coil (1)*.

### Effect of Corrected Stress

For springs of very large index (ratio of coil diameter to wire size) the stresses figured by these methods will not differ greatly; however, for springs of small or moderate index the corrected stress will be considerably higher than the uncorrected, the difference being nearly 60 per cent

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Numbers in parentheses refer to list of references at end of article.

for springs of index 3 and for springs of index 6 about 25 per cent (2). Thus in specifying working stress for springs it is important to state whether or not the corrected or uncorrected stress is to be used.

It should be emphasized that the stresses figured by either method (i.e., the corrected or uncorrected stress) will not, in general, exist in the spring at any given load. The reason for this is that, because of the usual cold-setting or scragging operations carried out by the spring manufacturer, residual stresses are set up in the spring which will be subtracted from the load stresses (3). Thus actual peak stresses are in general lower than the calculated. Although these methods do not, without modification, give an accurate picture of actual peak stresses, they do give an indication of the load-carrying ability of the spring under different operating conditions as will be shown later.

For purposes of choosing working stress, spring applications may be divided conveniently into three groups depending on the conditions of loading as follows:

1. Static loading (load steady or applied less than say 1000 times)
2. Intermediate loading (load applied say 1000 to 100,000 times during the life of the spring)
3. Fatigue or endurance loading (load applied $10^6$ times or more).

Each of these conditions requires a somewhat different design approach and method of stress calculation for best overall results, including spring performance and life.

### Steady or Infrequently Repeated Load

In the category of static loading are included those cases where the load is steady or applied less than, say, 1000 times during the life of the spring. Such applications include circuit-breaker springs where the breaker operates only a few times in its life, safety-valve springs where the valve pops off at rare intervals, springs for applying gasket pressure and the like. In such cases failure by fatigue fracture is not a problem, the chief concern being to avoid excessive set or creep. The choice of working stress thus depends to some extent on whether a small amount of set may be tolerated; if so, a higher working stress may be used than would be the case otherwise. The problem of creep or set is aggravated also by temperatures above 250 degrees Fahr. for ordinary steel springs.

At the present time, it is the consensus of most spring authorities that the stress augments due to curvature and direct shear may be neglected when calculating springs for static loading. In other words, it is believed that for static loading the *uncorrected stress* gives the better indication of the load-carrying ability of the spring. The son for this is that the corrected stress represents co tions within a localized region near the inside surfac the coil, while the uncorrected stress represents an aver stress over the entire surface of the spring wire or This average stress is more important as an index of sa where freedom from excessive set is required, as is usu the case for statically loaded springs.

Another justification for using the uncorrected stress a basis for calculation in statically loaded springs is

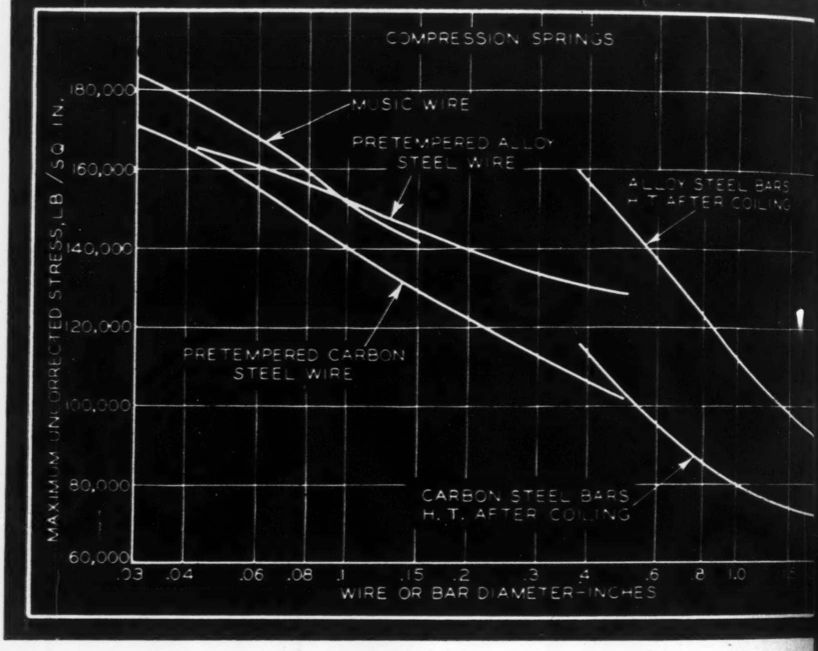


*Fig. 1—Stresses for compression springs in Ordnance applications where sp is limited. These stresses are maximum values and should be used only wh short life is required. Springs are assumed to be cold set and shot peened (f data published by S.A.E. War Engineering Board)*

following: If it be assumed that spring steel yields p tically at a definite shear stress equal to the shearing yi point, it will be found that the load for complete yieldi over the section (which will represent ultimate load fo perfectly plastic material with no strain hardening) be equal to 4/3 times the load computed by using the corrected stress formula and taking the stress equal to yield point (4). This assumes a spring of large or mo erate index. For smaller indexes, the effect of dire shear loading will modify this statement somewhat.

However, for most practical springs the effects of dire shear loading will be small and it appears satisfactory use the uncorrected stress. For small indexes, where co servative design is wanted, the uncorrected stress may multiplied by a shear stress multiplication factor whi takes into account the stress due to the direct-shear lo assumed to be uniformly distributed over the cross sectio In general, this method will yield results somewhat on t safe side (4).

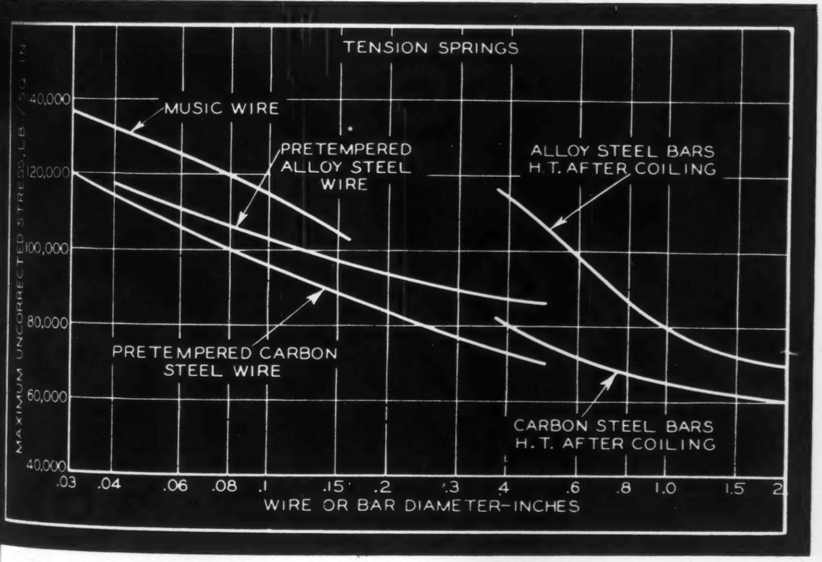
As an example of the extremely high working stress for helical springs which are used in practice, particular in ordnance design where space usually is at a premiu the values of *Fig.* 1 have been taken from data publish by the War Engineering Board of the Society of Autom

Engineers (5).

alues given in *Fig.* 1 apply to compression springs d primarily in ordnance applications where indefinite ing life is not required. These stress values are uncorted, i.e., calculated without curvature correction. In lying these stresses it is assumed also that the springs cold set and shot peened (for sizes over 1/16-inch) that the index is between 4 and 9. For index values ond these limits it is suggested that somewhat lower ss values be used.

imilar stress values for *tension springs* for ordnance apations based on data published by S.A.E. are given in 2. These values are in line with the usual practice aking lower stresses for tension springs than for comsion springs. This is done primarily since tension ings cannot be cold set without losing initial tension. values given for tension springs are approximately minimum expected values of the elastic limits in torn; for compression springs the working stress values y exceed the torsional elastic limit of the material but is compensated for by the effect of cold-setting operas during the manufacturing operations in producing spring.

It should be emphasized that the stress values of *Figs.* 1 d 2 are to be considered as maximum values which may used where necessary when space is at a premium and ere only limited life is expected. Where fatigue loadg is present (as in automotive valve springs) the use of ch stresses would generally result in early fatigue failure. r industrial applications where more space is usually ilable, it is the writer's opinion that somewhat lower esses than those given in *Figs.* 1 and 2 should be used erever possible even for static loading. In other words, values given in these figures are upper limits which y be used when necessary but which should be reduced erever possible.



***g. 2—Stresses for tension springs in Ordnance applications where space is ited. These stresses are maximum values and should be used only where ort life is required. Normally lower values should be used where possible (from S.A.E. War Engineering Board data)***

As a further example of working stress values, suggested *corrected stress* values for compression springs under occasional loading are given by Wallace Barnes Co. (6). These values for music wire are as follows:

| Diameter (in.) | Corrected Stress (psi) |
|---|---|
| .015 | 180,000 |
| .060 | 155,000 |
| .156 | 128,000 |

For average music-wire springs where the curvature-correction factor is about 1.2, these figures will correspond to stresses about 20 or 25 per cent lower than those in *Fig.* 1.

### Fatigue or Endurance Loading

Examination of springs which have failed from fatigue or repeated loading shows that, in most instances, the fracture starts at the inside of the coil at the point where the range of stress is a maximum. The stress range at inside of the coil may be calculated with sufficient accuracy by the use of the curvature correction factor (1). In view of these facts, it appears reasonable where fatigue loading is involved (i.e., where the spring is subject to a million or more load cycles) to calculate the stress range as a corrected stress.

Because of lack of sensitivity of some materials to stress concentration (as represented by curvature effects) it is probable that in most cases this method will yield results considerably on the safe side. In other words, for low-index springs the actual endurance range calculated by curvature correction may be considerably above the endurance range obtained by tests on large-index springs of the same material and wire diameters. When more test data are available this effect may be taken into account by using a so-called "sensitivity index". This index will depend on material and may be expected to be rather low for hot-wound heavy springs with a decarburized surface and higher for high-grade steels such as are used in automotive valve springs.

In the absence of actual test data, however, a conservative method of design is to use the full correction factor for calculating stress range where fatigue conditions are involved. This stress range may be compared with the results of fatigue tests on springs in order to get an idea of the factor of safety. In doing this, it should be considered that for smaller-index springs higher endurance ranges will probably be found than for larger-index springs on the basis of available test data. Also the endurance range will be somewhat reduced when the spring is under a high mean stress as compared to that obtained for a zero to maximum type of stress application, and this should be considered in design.

Values of corrected stress ranges as used in automobile valve springs by different manufacturers are given by Zimmerli (7). The stress range in these spring applications usually is between 30,000 and 40,000 pounds per square inch for carbon-steel springs, while the peak corrected stress of the range varies from 60,000 to about 85,000 for different designs. This range is roughly one-half to two-thirds the endurance range of the material, which would correspond to a factor of safety of around 1.5 to 2. Because of surging and vibration in valve springs, however, the actual stress ranges will be somewhat higher than the figures mentioned, and this would tend to reduce the margin of safety.

### Intermediate Loading

Perhaps the great majority of spring applications fall in the category of intermediate loading which includes springs subject to the peak load 1000 to 100,000 cycles during their lives. Such springs would be classed under a more severe application than where only static loading is involved but not as severe as fatigue loading. Examples of intermediate loading applications are suspension springs for automotive and military vehicles, springs used in "cup" drives for electric locomotives, switchgear and circuit-breaker applications involving frequent operation, etc.

An estimate of the amount of increase in allowable stress where springs are subject to 100,000 or fewer cycles of load application may be observed by using the fatigue test results reported by Edgerton (8). These test results, shown in *Fig.* 3, were obtained on carbon-steel springs (¾-inch bar diameter, index 5) tested in a zero to maximum stress range. In this curve the corrected-stress range is plotted against number of cycles to failure (log scale). Although there is considerable scatter the trend of the average is indicated by the full line. From this it is seen that for a life of 100,000 cycles the allowable stress is about 1.5 times the stress for a life of one million cycles. For centerless-ground bar springs shot-peened, of index 5, higher values would be expected. However, for springs of larger index somewhat lower values may also be obtained.

Complete agreement as to the proper method of desi ing springs for applications involving a limited number cycles does not exist among authorities at present. Th is a difference of opinion as to whether or not curvatu correction should be used in figuring the stress range. though further research will be needed to settle this qu tion, a conservative method of design is as follows:

The stress at maximum load should be figured as unc rected stress (i.e. without curvature correction), a should not exceed the allowable value for static loadi (see previous discussion). The stress range, obtained fr the upper and lower limits of the load range, should figured with curvature correction and this should be co pared with the endurance range for the required numb of cycles using curves of the type shown in *Fig.* 3. F small numbers of cycles and for small indexes, i.e., larg curvature-correction factors, this method will probab err considerably on the safe side, since as the number cycles is reduced the condition tends to approach a sta load condition where curvature correction may be negle ed. Further test data are needed to throw light on t question.

As an example of working-stress figures suggested average service conditions the values given in Table I

TABLE I

**Recommended Maximum Design Stresses in Shear for Helical Compression and Extension Springs Under Average Service Conditions***

| Wire Diameter (inches) | Kind of Wire (Plain Carbon Steels) Music (psi) | Tempered (psi) | Hard Dr (psi) |
|---|---|---|---|
| .020 to .030 | 100,000 | 100,000 | 90,00 |
| .031 to .092 | 90,000 | 100,000 | 80,00 |
| .093 to .176 | 90,000 | 90,000 | 80,00 |
| .177 to .282 | ...... | 90,000 | 70,00 |
| .283 to .436 | ...... | 85,000 | ..... |
| .437 to .624 | ...... | 80,000† | ..... |
| | ...... | 90,000†† | ..... |
| .625 to .874 | ...... | 90,000‡ | ..... |
| .875 to 1.500 | ...... | 80,000‡ | ..... |

*Table based on data published by American Steel & Wire Co. Cu ture correction included.

†Cold wound for O.D. greater than 3 inches, otherwise hot wo

††Hot wound for O.D. less than 3 inches.

‡Hot wound.

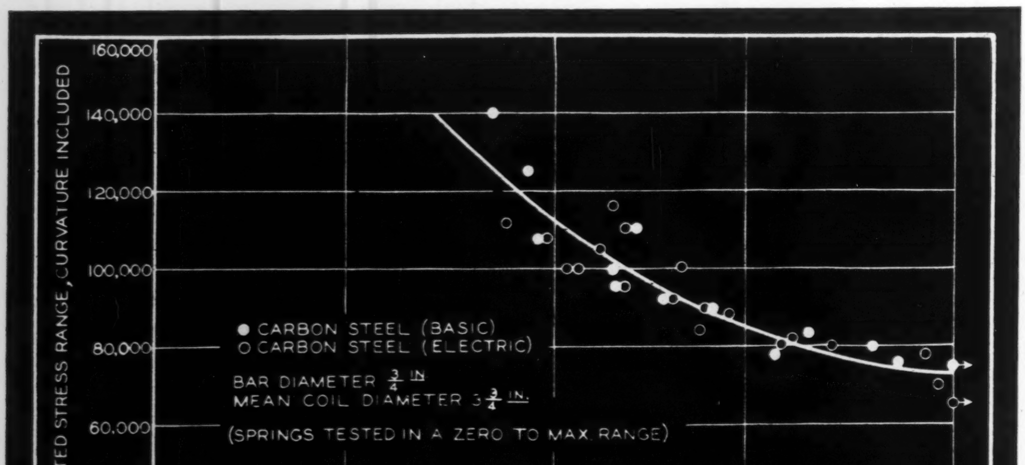


*Fig. 3 — Results fatigue tests of h wound heavy heli springs (based on te by Edgerton, Transa tions A.S.M.E., O 1937 and Progress R port No. 3 of A.S.M Spring Committee*

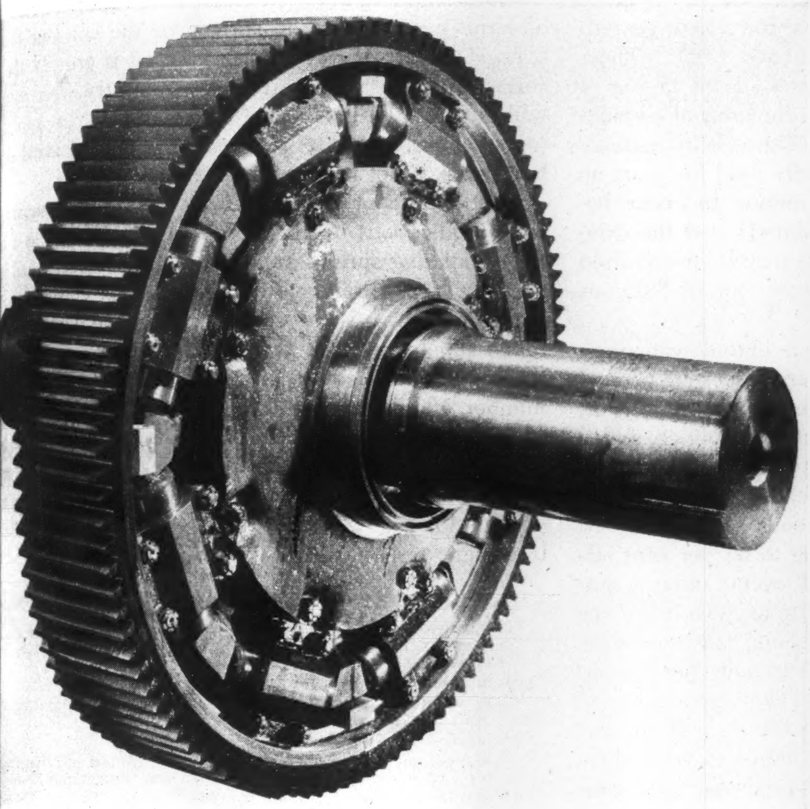


Fig. 4—Left—Cup drive for steam turbine locomotive

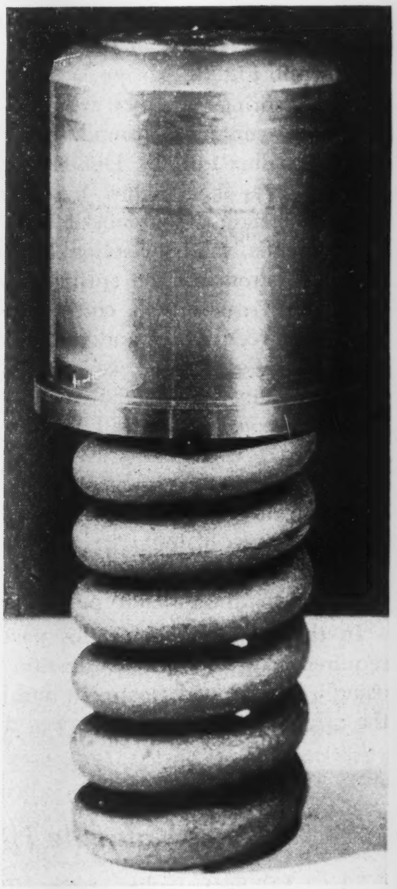


Fig. 5—Spring for locomotive cup drive shown in Fig. 4

ed on data published by American Steel and Wire (9).

verage service conditions in this case (TABLE I) are ned as noncorrosive atmosphere, temperatures not over 0 degrees Fahr. and under relatively slowly varying or tic loads. Since curvature correction is included in the ss figures of TABLE I, it will be noted that for average ng indexes where curvature correction is about 20 per t, these values are equivalent to lower uncorrected sses than those of *Fig.* 1 for ordnance applications.

### Application to Problems

pplications of some of these principles in design will illustrated by the following examples taken from actual ctice.

XAMPLE 1: As an illustration of an intermediate load- application, a spring used in a circuit breaker is de- ned for a life of 100,000 cycles or more. The function this spring is to apply pressure on the contacts when the ker is closed and also to accelerate the contacts on ping the breaker. Dimensions are: Outside diameter nch, wire diameter .071-inch (music wire), index 7.8, tive turns, free length 2¼ inches, closed length 1 3/32 hes.

n operation the load varies from a minimum of 11.7 nds to a maximum of 27.3. The uncorrected stress at imum load is 108,000 pounds per square inch and the *ected stress* range between minimum and maximum d during operation is 73,000 pounds per square inch. ay be seen that in this case the working stress used considerably lower than the value given in *Fig.* 1 for 1-inch wire. This is typical of practice in switchgear and circuit-breaker applications where working stresses appreciably lower than those of *Figs.* 1 and 2 are generally used.

EXAMPLE 2: An example of an application of a tension spring in circuit-breaker design, is given by the following spring, which is designed for a life of 50,000 cycles or more. Dimensions are: Outside diameter 9/16-inch, wire size .102-inch (music wire), index 4.5. Spring ends, screwed into ends of the spring, are designed so that at low loads there are 6½ active turns and at higher loads, 8½ active turns.

In operation the load on the spring varies as follows: 15.5 pounds minimum with breaker in trip position, 53.5 pounds with breaker in open position and 106.7 pounds maximum load during closing operation. Uncorrected stress at maximum load 118,000 pounds per square inch. In 99 per cent of the operations load range is from 53.5 to 106.7 pounds corresponding to a corrected stress range of 80,000 pounds per square inch. This range of stress corresponds to an expected life, based on fatigue tests results, of over 10 times the required life.

EXAMPLE 3: An interesting example of the design of heavy springs for intermediate-loading conditions is given by the springs used in the "cup" drive for a new geared-turbine locomotive described in reference (10). These compression springs are inside the cups shown in *Fig.* 4 and form the flexible element between the arms of a spider pressed on the axle shaft and the gear driven by a steam turbine. This type of drive, successfully used for years on electric locomotives, allows relative motion to occur between the gear (which is spring mounted) and the driving axle while at the same time torque may be transmitted to the axle. A view of the spring inside one of the cups is shown in *Fig.* 5.

These springs are hot wound from chrome-vanadium steel bars, centerless-ground. They are also shot peened and magnaflux tested. Dimensions are as follows: Outside diameter 3.56 inches, bar diameter, .870-inch, index 3.1, free height 10.51 inches, 8¾ active turns, solid load 7650 pounds, solid deflection 1.59 inches. Maximum uncorrected stress on the spring with coils solid is 80,000 pounds per square inch, corresponding to 57 per cent adhesion between wheel and rail, which occurs only at rare intervals. For normal starting conditions where 30 per cent maximum adhesion between wheel and rail is assumed the uncorrected stress will be 42,000 pounds per square inch and the corrected stress 65,000. This stress range of 65,000 pounds per square inch is less than half the expected endurance range (8) for 100,000 cycles for shot-peened springs of index 3. Under normal operating conditions with 18 per cent adhesion the corrected stress in this spring will be 39,000 pounds per square inch.

In the manufacture of this particular spring care was required to avoid decarburization of the bar surface by using a special heat treatment and by grinding off some of the original bar stock. This was done since there is considerable rubbing between the outside surface of the sp and the hardened inside surface of the cup due to trifugal force. If any decarburization is present, the surface material will be rubbed off and the wear prod will get into the lubricating system. For most spring plications of this type, such extreme care to avoid de burization is not required, however.

On the basis of available data, it appears reasonable most applications to use the uncorrected stress as a b for calculating springs subject to static or occasional lo ing only. For springs subject to fatigue or endurance lo ing, at present it is the consensus that the corrected st range should be used as a basis for design. For spri subject to intermediate loading conditions with a limi number of load cycles, it is suggested that for conse tive design, the maximum load be calculated on the b of the uncorrected stress while the stress range be cal lated using curvature correction. This range may be co pared with the endurance range for limited life obtai by fatigue tests.

### References

(1) Further discussion of methods of calculation of helical sprin given in *Mechanical Springs* by the author, published by P Publishing Co., 1944.

(2) *Mechanical Springs*, Page 37.

(3) For methods of calculating residual stresses, see author's ar MACHINE DESIGN, Jan. 1945, Page 107.

(4) *Mechanical Springs*, Pages 100 and 105.

(5) *Manual on Design and Application of Helical and Spiral Sp for Ordnance*, published by S.A.E. War Engineering Board.

(6) *Mechanical Springs, Their Engineering and Design*, published Wallace Barnes Co. Div. of Assoc. Spring Corp., 1944.

(7) "Permissible Stress Range for Small Helical Springs", Unive of Michigan *Engineering Research Bulletin* No. 26.

(8) *Progress Report No. 3 on Heavy Helical Springs* by A.S. Special Research Committee on Mechanical Springs. Abstract C. T. Edgerton, *Transactions A.S.M.E.* Oct., 1937, Page 609.

(9) *Manual of Spring Engineering*, 1941, published by Ameri Steel & Wire Co.

(10) "How the Geared-Turbine Locomotive Works"—Newton Brecht, *Westinghouse Engineer*, March 1945, Page 37.

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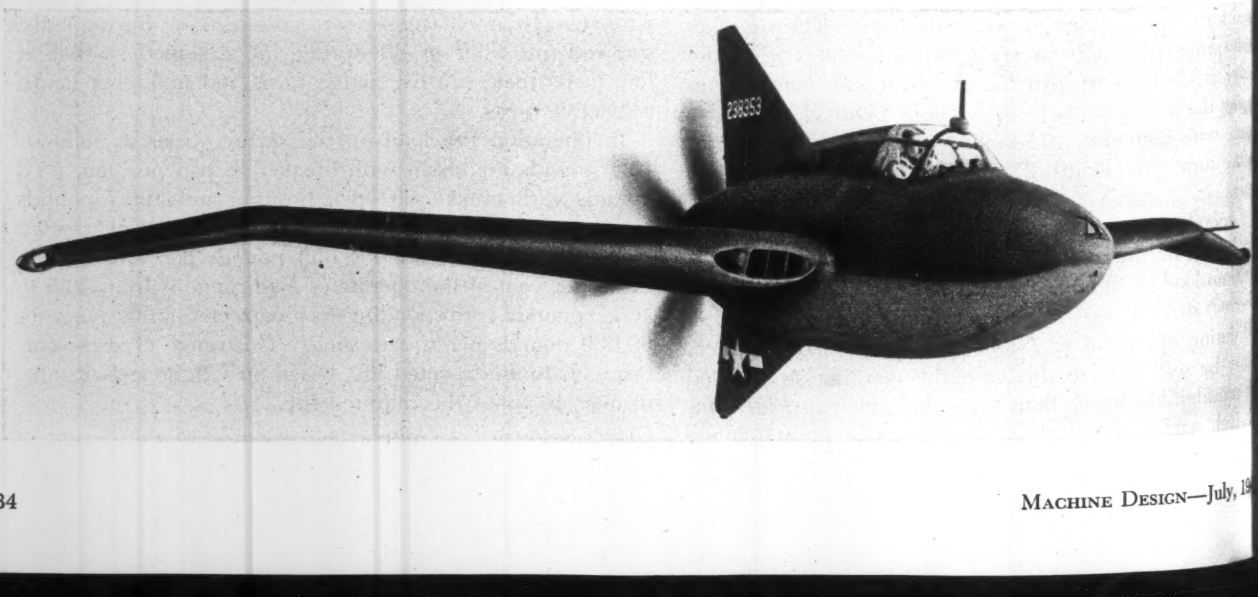
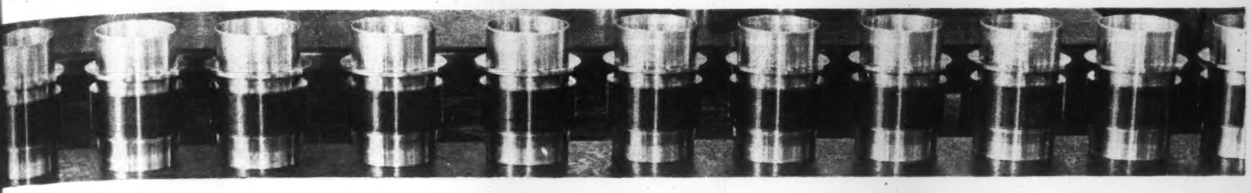
## *Gives Valuable Flight Data*

SUCCESSFUL flights of the XP-56, Northrop's pusher type pursuit plane, have yielded valuable research data which it is believed will materially advance aircraft developments. Technically a tailless plane carrying pilot and blower-cooled engine in a streamlined nacelle, it is the first of its kind ever designed. A radial air-cooled engine drives two three-bladed counter rotating pusher propellers, giving thrust without torque. Elevators and lateral controls are on the wing trailing edge, while fi vertical fins above and below the fuselage provide di tional stability.

This is the first plane of its type with a complet submerged air-cooled engine. The streamlined nace vertical fins above and below, and the absence of a c ventional tail give the plane a startling appearance. T flights, beginning in the latter part of 1943 and conti ing up to the present, have afforded unusual techni information for military and peacetime aircraft designs.





# Statistical Control—the Yardstick of Performance

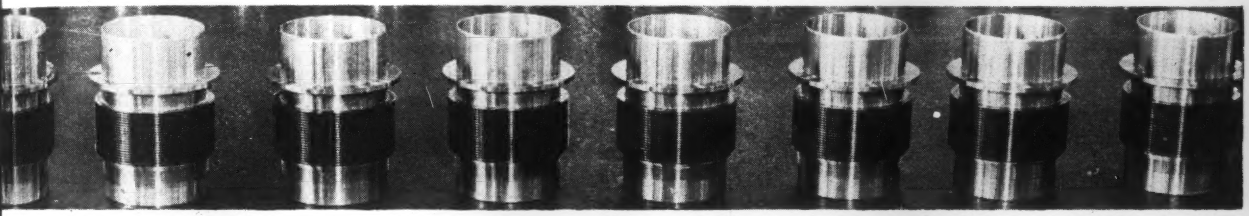


*Fig. 1—Mass-produced precision parts must conform to rigid specifications. Photo, courtesy Buick Motor Div., General Motors Corp.*

**By Roger W. Bolz**

***Assistant Editor, Machine Design***

STATISTICAL methods of investigation and control provide the fundamentals for what may well be called the quality gage of a production achine. The potentialities of this tool offer much ward real improvement in such machines. As now apted and applied to the vast quantity of vital r material produced in government arsenals[1] and merous other manufacturing plants[2], it has proved value beyond any question, whether we are inested in inspection and conformance of products in the control of the process used in the manucture of these products.

The latter of these two factors, namely, knowledge the various characteristics of a specified producn process, and control of those characteristics, e of primary interest to the machine designer. It well known in manufacturing circles that all too ten a process seems to be "haywire" or appears imssible to continue economically without a conderable relaxation in tolerances. This problem can st easily be avoided by a statistical analysis of:

1. Tolerances required in the parts produced
2. Physical characteristics necessary in the parts
3. Characteristics of the processing method.

om these facts it can be determined whether or not the process can produce the parts to suit specifications at a reasonable cost. The question as to whether the problem rests with men, materials, or machines—a vexing, complicated and involved problem—can be narrowed down to a field small enough to justify a thorough investigation.

No matter what particular production process may be under consideration, the machine or machines involved in carrying on this process must operate with a uniform or constant level of output. Without the use of a statistical test procedure, the overall output characteristics of a production machine and more so of a series of machines can only be surmised. The wide field of possibilities for developing and improving the quality of the machine and ultimately the quality of the product produced by it has not been adequately examined.

Any manufacturer using a production machine necessarily needs to know that the process involved is dependable and that no unforetold changes occur during the operating cycle. He must be sure that he can and actually does produce the quality intended. Statistical tests will meet these demands and at the same time save much time, labor and material which otherwise would be lost during the attempt to control output by mere inspection procedure during production. The fact that one part or even a quantity sample of many parts at various intervals check satisfactorily by individual gaging does not in any way present a true picture of the actual quality level of machine performance. It has been definitely proved that such random samples repeatedly indicate a quality level considerably above that of the parent lot Inspection of this nature

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[1] "Quality Control in Manufacture of Small-Arms Ammunition"- M. Smallwood, *Mechanical Engineering*, March, 1944.

[2] "Consumer Control of Quality by Inspection", *Metal Progress*, , 1944.

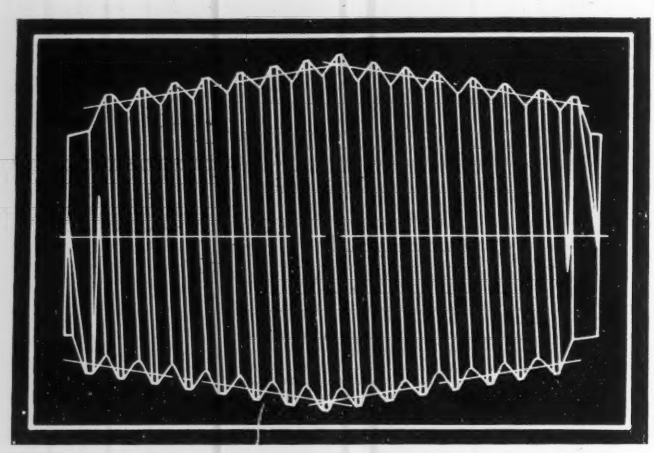
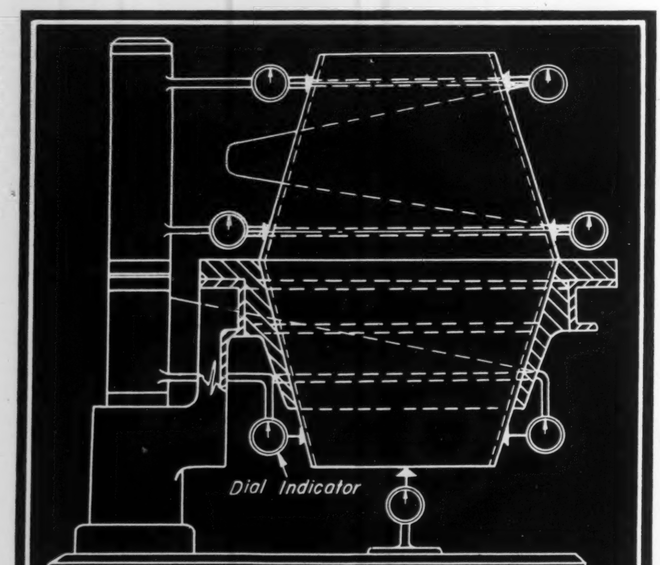


*Fig. 2—Above—A special double taper threaded part produced in two operations on an automatic thread miller*

*Fig. 3—Below—Combination gaging arrangement for obtaining all variables at one reading of dial indicators*



used to establish the conformance to requirements and the effectiveness of the process, therefore, is ruled out—control of the basic process or machining functions remains as the best means for obtaining effective and uniform product quality.

One authority[3] states, "the control chart is the only satisfactory solution to the standardization of machine performance." Controlled or standardized performance is essential. Regardless of whether one machine or many machines are involved, proper application of the statistical procedure may well disclose some unexpected or unusual characteristics in the processing. During the development stages of a machine, statistical control charts permit the machine designer to observe the trend of deviations from the set dimensions of the parts being manufactured. The chart of continuous production operat lays before him a "case history." From this case hist he readily may detect discrepancies in output. Va tions or trends in sustained repetitive operation are mediately visible. There is no need to guess at the put quality characteristics because the status is bef him clearly pictured. The machine is either under statistical control, producing at the maximum quality l el possible, or it shows signs of unpredictable variati and evidences by means of the statistical chart a defi lack of control.

Where a specific maintained dimensional accuracy output is required, the chart may show that only 50 cent of the parts produced can be guaranteed to c form. Is redesign necessary to enable fulfillment of specifications or will the elimination of one or more signable causes correct the trouble? The case hist of machine performance presented by the control ch will point the way to rational steps in the further de opment of the machine in question.

## Sound Basis for Redesign

The distribution of dimensions indicated by a con test forms a sound basis for changes in the method processing or machine design necessary to obtain the quired quality level. At no time will the dimensi accuracy of mass-produced parts, *Fig.* 1, be perfect. chine performance from hour to hour and over extend periods of production always varies. However, the important factor to consider is the magnitude of the v ation, or in other words, the distribution of dimensi during such periods of continuous production. If distribution of dimensions proves to be stable and c stant in its range, the machine usually can considered as operating under a controlled proce Improvement in this output quality or reduction the range of variation usually will require an alterat in the fundamental process, or redesign of the machine.

An interesting problem in machine performance presented by the special taper threaded part shown in F 2 which was being produced on a special set of thr milling machines. This part involved extremely cl tolerances on relative pitch diameters to enable lock the faces of the female mating pieces and at the sa time maintain maximum threaded engagement. Ma facturing and assembly problems proved ordinary go no-go type ring gages were inadequate. Precise dial indic tor measurements made with a combination gage unit signed to show all deviations, *Fig.* 3, at one reading p vided the basis for statistical analysis.[4] The first cha obtained during production brought to light a disti lack of stability in continuous output. Apparent in chart reproduced in *Fig.* 4 is this lack of uniformity dimensions of samples taken at exact intervals over period of days. Spotty conditions are evident, with taper of the threaded part continually shifting from side of the mean taper to the other. The great perc tage of points that fall on or outside of the maxim limits indicates a need for action to locate the cause the extreme variation. Pitch diameter chart of the sa products in *Fig.* 5 indicates a similar condition.

A study of the machine and process finally showed lue of the indications in the charts. After checking each mple it was found necessary, in most cases, for the oper- or to shift the cutting head of the machine for what peared to be excessive cutter wear. However, cutter ear proved to be only a minor consideration and the in- cations in the chart therefore showed that shift of di- ensional size was primarily in one direction following ch reset. Further tests indicated that the pitch diam- er variations were, in part, a direct result of the taper anges. Inspection of the design features of the cut- g head of the machine showed that the clamping echanism utilized to retain the head in preset position was far too fragile to resist the loads imposed upon the head by the cutter. Clamping the head merely produced a bending action in the mechanism, frictional holding power developed by the clamp being very small.

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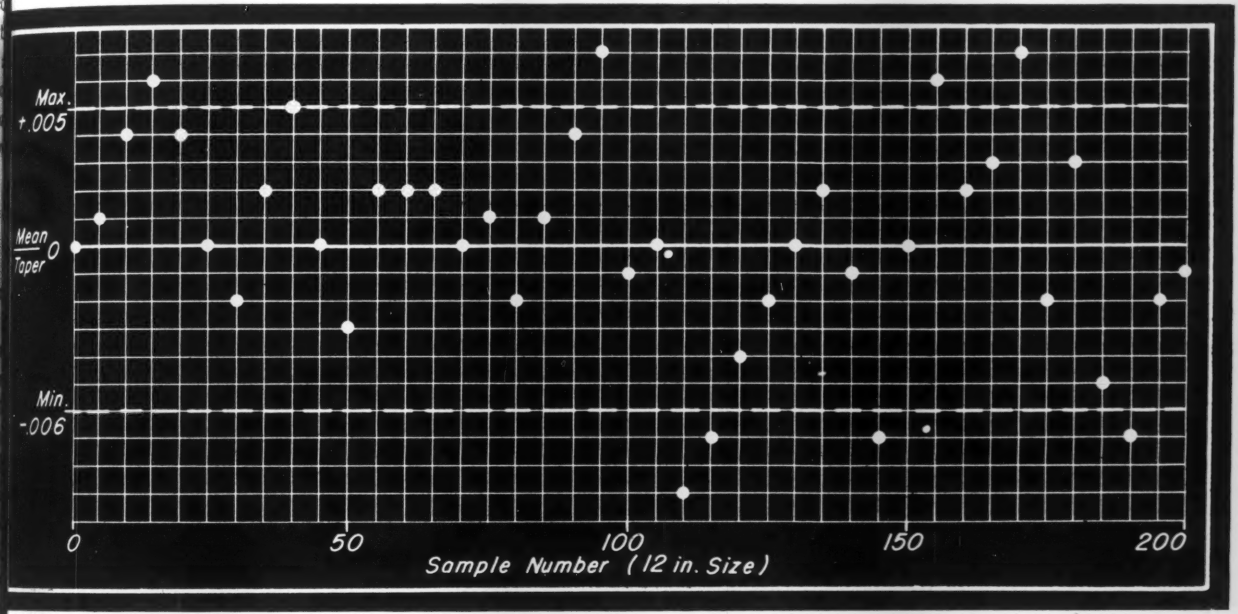
[3] W. Edwards Deming—"Some Principles of the Shewart Methods of Quality Control", *Mechanical Engineering*, March, 1944.

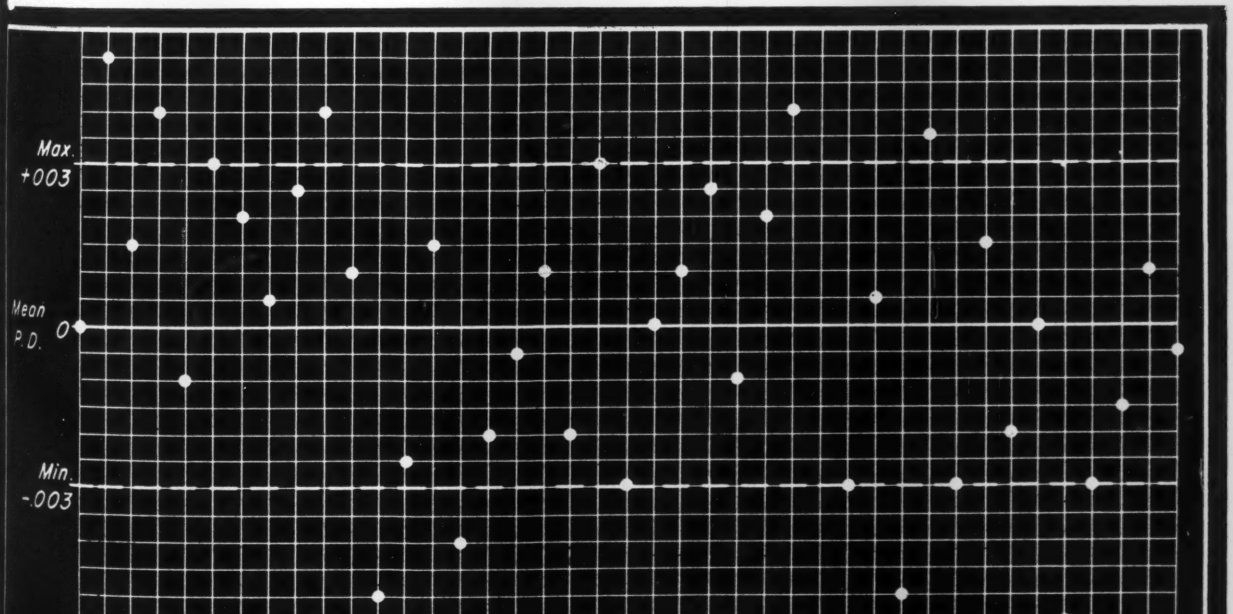
[4] Quality sampling by variables is considered the most efficient procedure—*An Engineers' Manual of Statistical Methods* by Leslie E. Simon, Wiley, New York, 1941.

This condition was remedied easily by redesigning the entire clamping arrangement. Resultant production figures were improved decidedly. The charts in *Figs.* 6 and 7 indicate the stabilized output obtained by this redesign. All points lie safely within the control limits and no widely scattered patterns are present. Dimensional output of the machine now appears to be constant and all tapers fall within plus or minus .003-inch and all pitch



***g. 4—Above—Chart showing lack of control in taper of special dauble taper threaded part***

***Fig. 5—Below—Pitch diameter spread indicates similar uncontrolled variations in size of the samples in Fig. 4***



diameters within plus or minus .0015-inch. It can be concluded that all assignable causes of variation have been eliminated. However, if specifications had been even more exacting and the control limits necessarily set to conform to these tolerances on the part, a fundamental design change would have had to be made. The machine could not be expected to operate within a smaller spread than that indicated by the statistical data obtained under normal operating conditions.

The manufacturing range indicated by the latest charts, *Figs.* 6 and 7, in inherent in this particular process used and would, under the more exacting specifications, be much greater than the tolerance range. There would be no way to meet, with any degree of assurance, these new limits except by instituting a 100 per cent inspection. This would be both costly and subject to human failings from sheer monotony of procedure. Where specified tolerances are exceedingly close, even such critical influences as changes in temperature and variations in the physical characteristics of raw materials necessarily would have to be taken into account in order to avoid expensive process changes or adoption of more liberal dimensional limits.

Inasmuch as any production machine will operate with a continuous output that varies in dimensional quality, the important problem is not to seek to eliminate va tions but to stabilize and control the spread of dimens al output over extended production periods. This, t is the true indication of overall performance which e lishes the statistical control method as the real qu gage of a production machine. It is not limited to l manufacturers nor to machines produced in great volu Equally valuable results are possible whether a g many identical machines or only a few models of a spe type machine are being built. The need is much gre where only one or two special machines are made. In cost is high and performance demands are often m more critical in such cases, and an analysis of the prod tion performance of the first model should be made bef building a second.

Design requirements specify a quality goal which to be met. Engineering knowledge brought to light statistical quality control procedure definitely indic from an economic point of view whether this goal can met. At the same time it reveals in a process those v ables which may be left safely to chance and those whi for improvement or stabilization in quality and savings time, labor and materials, should be tracked down eliminated as soon as practicable.

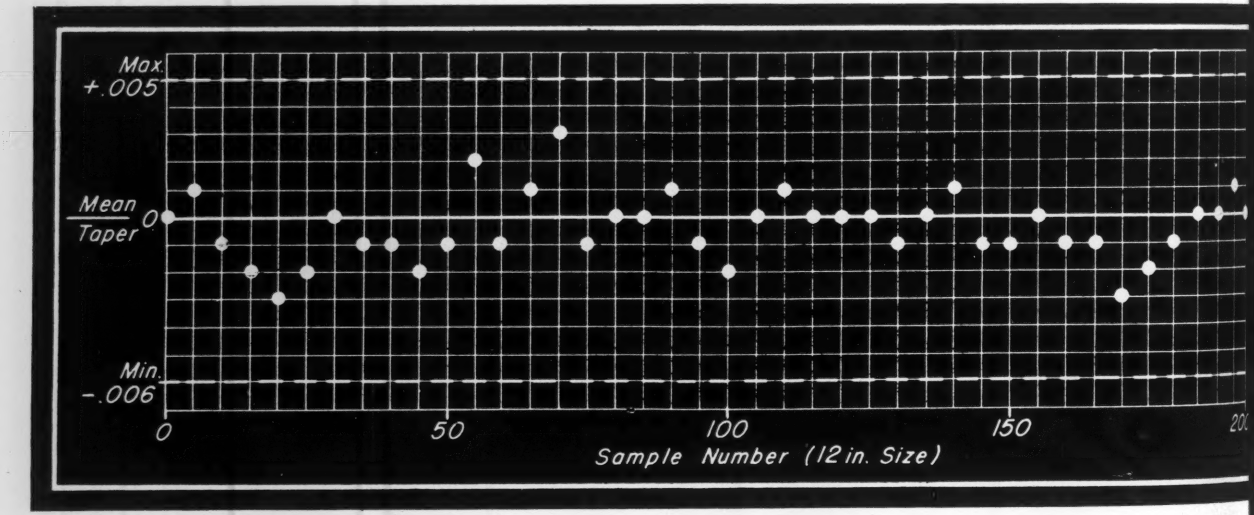
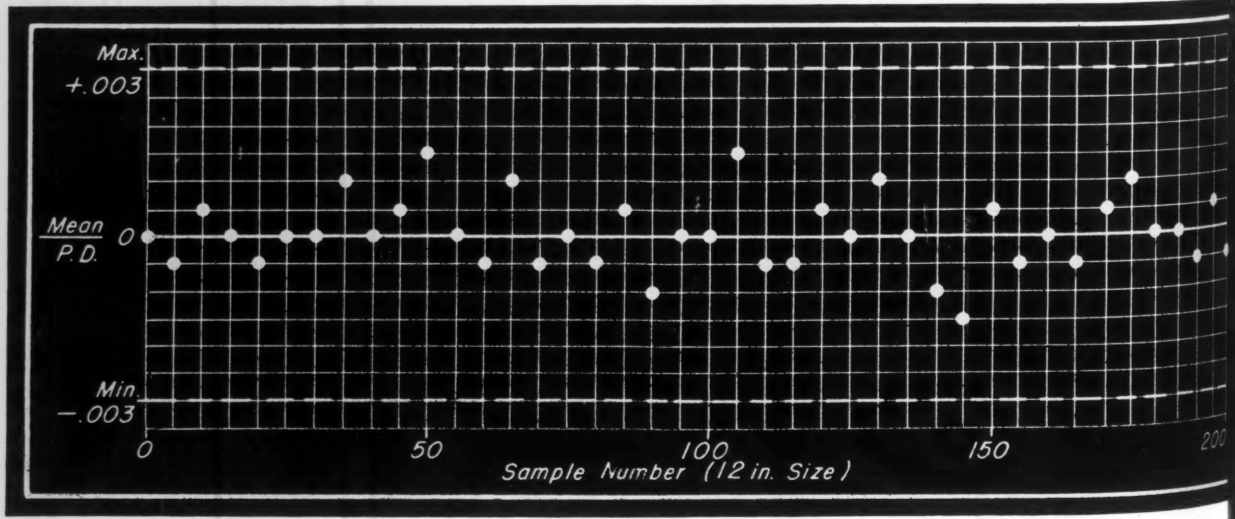


***Fig. 6—Above—Output improvement after correcting faults in clamping mechanism of machine***

***Fig. 7—Below—Pitch diameter spread also shows m improvement and good control in the samples of Fi***



# MACHINE *Editorial* DESIGN

## New Standard Should Be Welcomed

*MANY are the instances that could be cited wherein war production has been speeded up solely by means of uniform practice. And many are the instances that have arisen in which delay and confusion have occurred due to lack of established standards in certain lines.*

*Well remembered in this connection will be the early attempts to produce Rolls-Royce aircraft engines in this country to drawings furnished by the English company. So many discrepancies were brought to light between the drafting standards, tolerances, etc., on which these drawings were based and those in use here, that months of work were necessary in making over the drawings to conform to American standards. Had international standardization been further advanced, a great deal of time and effort could have been saved and utilized in more effective work.*

*International standards obviously cannot be established until our own house is in order—a fact well recognized by the American Standards association. Much credit is due that organization for the promulgation of the current standard on drafting room practices, which incidentally might well be classified as a "grass roots" standard without which engineering standardization in general cannot readily succeed. That the present standard covering engineering department work is not sufficiently comprehensive, however, is evidenced by the fact that the military and naval services have agreed jointly to back the rushing to completion of a more all-engrossing standard on drafting room practices to fill their present and possible future needs as well as those of industry.*

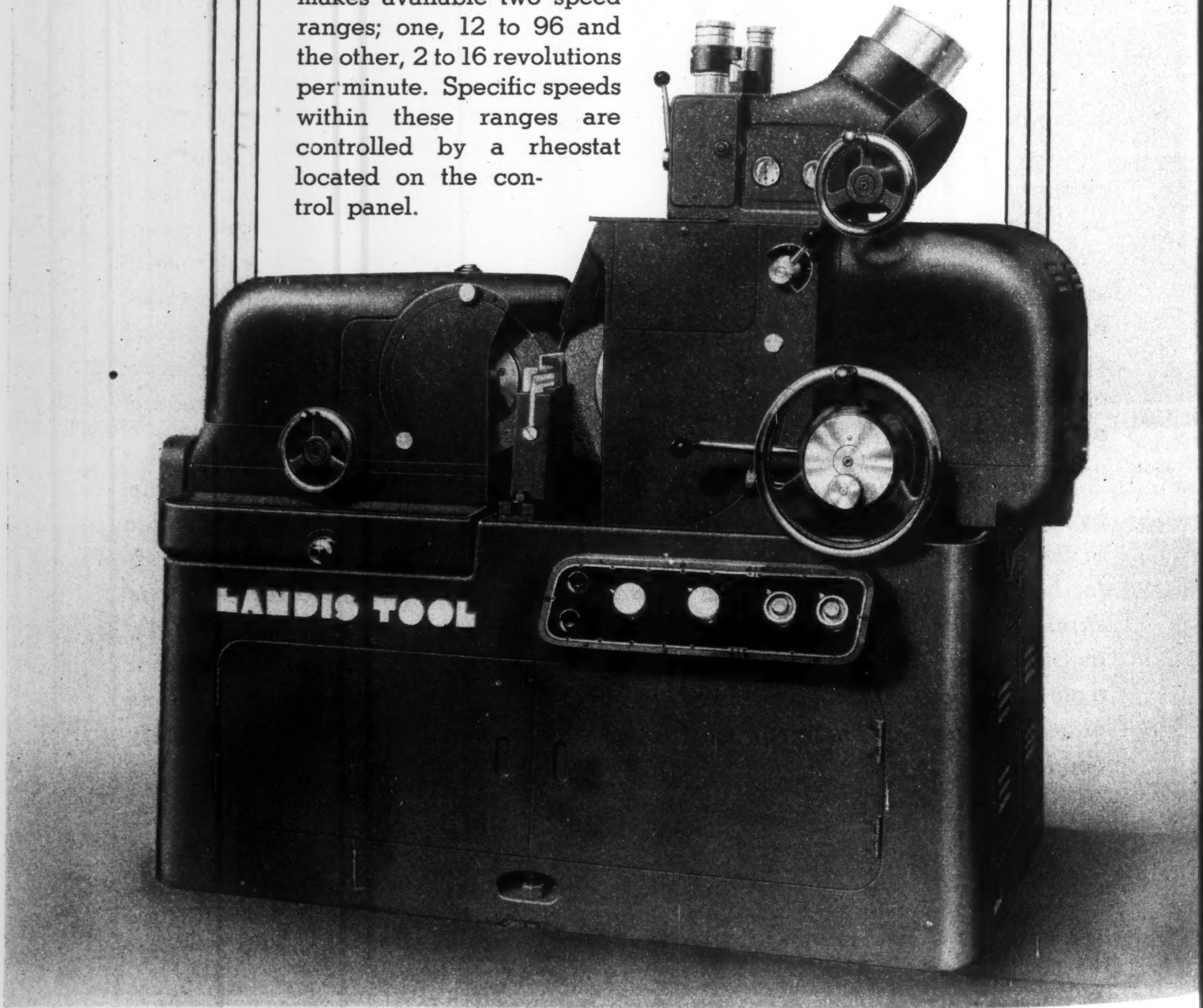
*Such a standard should be accepted as wholeheartedly as possible by industrial companies, not only for the assistance it may be able to render in the war but also for its peacetime implications. Given the much-hoped-for boom in trade and the trend toward modernized design after peace is signed, skilled help is likely to be at a premium in engineering departments, as at present, throughout the country. It is under such circumstances that a clean-cut, comprehensive standard covering all the pertinent ramifications of engineering department work could, and should, be put to extremely good use.*

L. E. Jermy

# Outstanding Designs

## Centerless Thread Grinder

To permit grinding of screw threads, this centerless grinder manufactured by Landis Tool Company employs a regulating wheel adjustable by handwheel in both vertical and horizontal planes as well as transversely. A "crusher" is used to roll the desired thread form or profile into the grinding wheel face, being driven through a worm gear reduction by a constant-speed motor. The crusher is a hardened steel cylinder with accurate annular grooves, mounted on its shaft in such manner as to permit ready replacement. Base of the regulating wheel (left) houses a clutch mechanism that makes available two speed ranges; one, 12 to 96 and the other, 2 to 16 revolutions per minute. Specific speeds within these ranges are controlled by a rheostat located on the control panel.

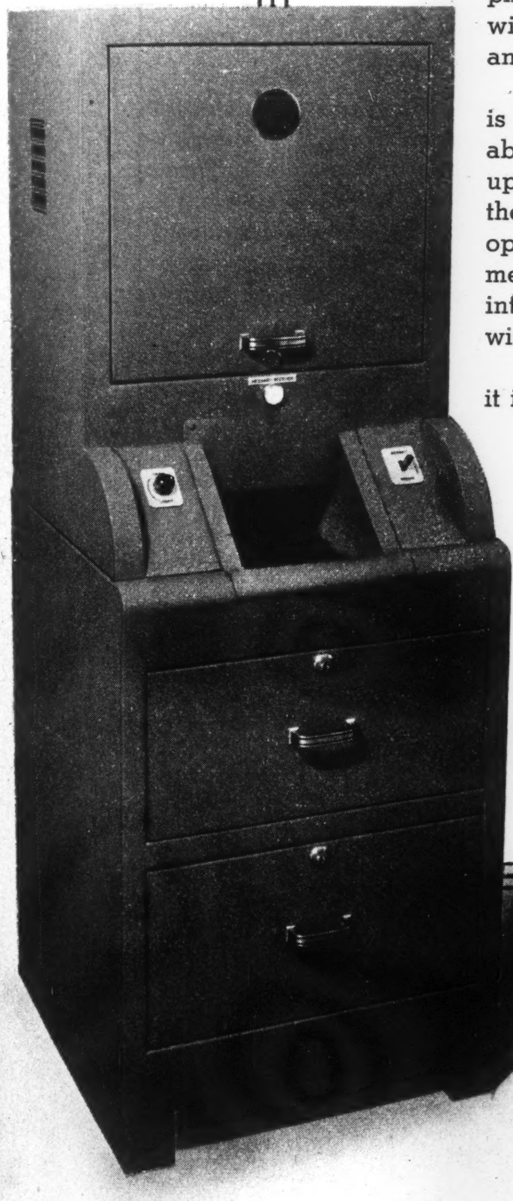


# Automatic Telegraph Recorder

Recently developed by Western Union Telegraph Co., this facsimile recorder is loaded with 400 blanks of dry recording paper in much the same manner as a camera is loaded with a roll of film. A small motor, photoelectrically controlled, automatically withdraws blanks from the roll as needed and places them in position for recording.

An electromagnetically operated roller is employed to wrap the recording blank about the recording drum by impaling it upon a ring of spiked projections around the circumference of the drum. This operation takes place at the instant the message to be transmitted is deposited into the "mail box" type of transmitter with which the recorder is associated.

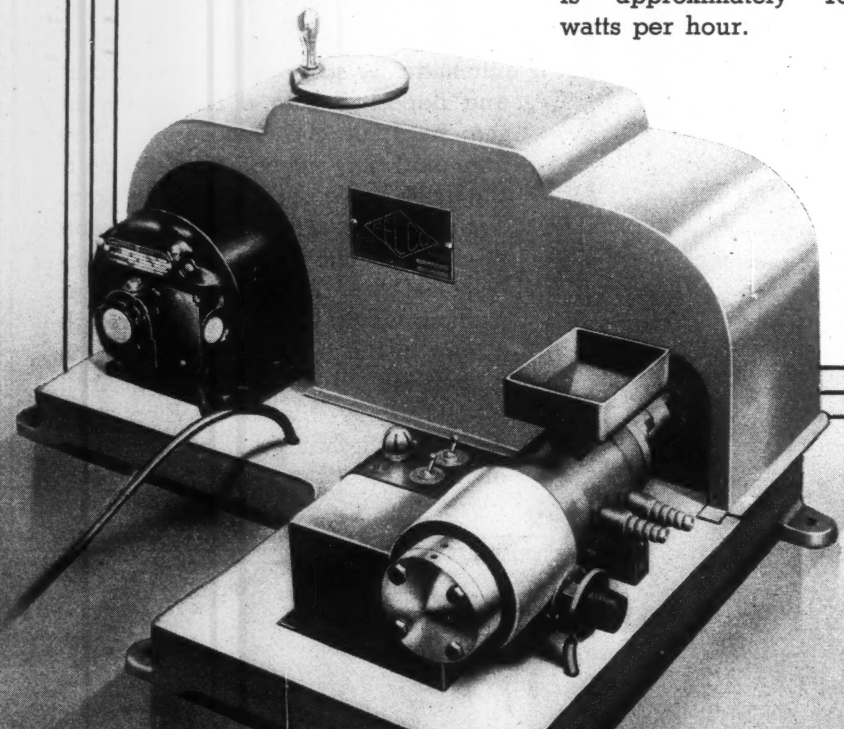
After the message has been recorded it is automatically stripped from the drum and deposited into a receptacle in the front of the recorder. The attendant may accept it or secure a duplicate copy by pressing a key. The recorder is housed in a green crinkle-finished cabinet with stainless steel trim. Part of the mechanical portion, which is accessible through a door in the front of the cabinet, is hinged to facilitate replacing the roll of recording blanks. Electronic equipment is housed in two drawers in the base of the cabinet.



# Laboratory Plastics Extruder

Used primarily for laboratory extrusion of plastic compounds, this machine, made by Production Machine Co., is all electric and completely self contained. Its 750-watt electrical heating element, controlled automatically by a thermo-switch installed in the heating chamber, provides for a heat range up to 600 degrees Fahr., controllable within five degrees.

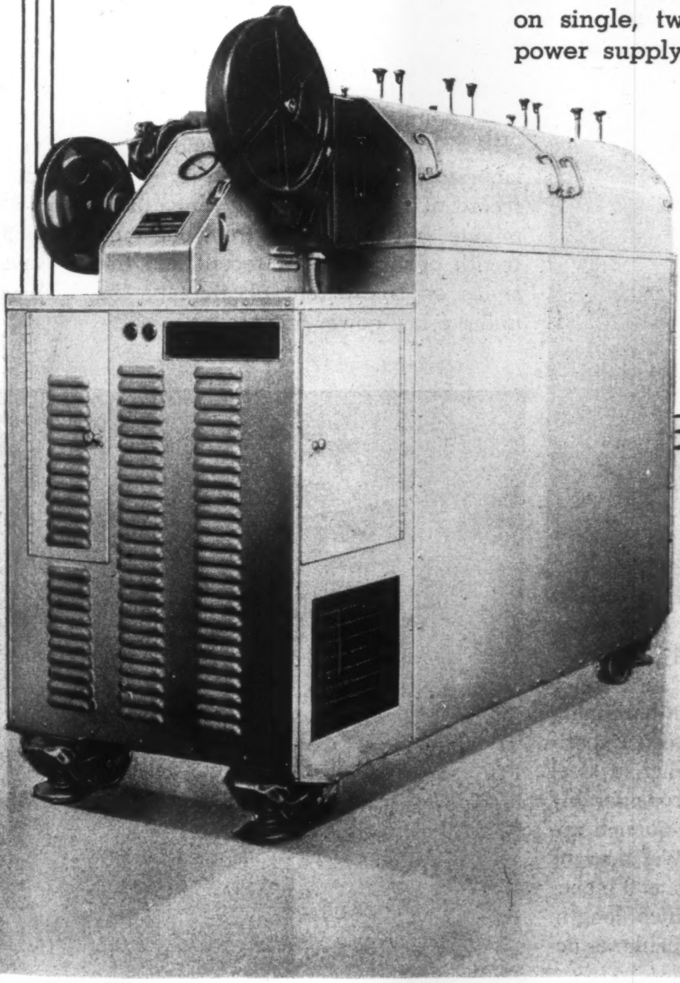
Operating in a hardened steel liner, a chrome-plated, polished screw having a compression ratio of 12:1 develops a pressure more than ample for the requirements. The machine is equipped with a variable-speed drive allowing a screw speed of 5 to 20 revolutions per minute. Higher speeds may be obtained by changing the motor pulley. Total power required to operate the extruder at high heat is approximately 1000 watts per hour.



# Film Processor

This machine, manufactured by The Houston Corporation, develops negative, positive and reversal 16-mm films. Its base is constructed of three-inch channel iron and 10-gage steel plate. Water jacket, solutions tanks, superstructure and power compartment, as well as all shafts, pipes, fittings, brackets, bearings, turbulence nozzles and rods that come into contact with chemical solutions are of stainless steel. Four magazines with loading flanges, loading bracket and the take-off housing are aluminum.

Thermostatic control of the refrigerator coils and the heater within the water jacket maintains the solutions at a constant temperature. Bakelite spools or rollers are used for supporting the film throughout. A control panel, including a start-stop switch, pilot light for flashing lamps, thermometer for solutions, a speed indicator and footage counter, are placed conveniently for the operator. All wiring is in conduit, arranged so as to allow the unit to be operated on single, two or three-phase power supply without need of any internal changes. Drying is by means of infra-red lamps.

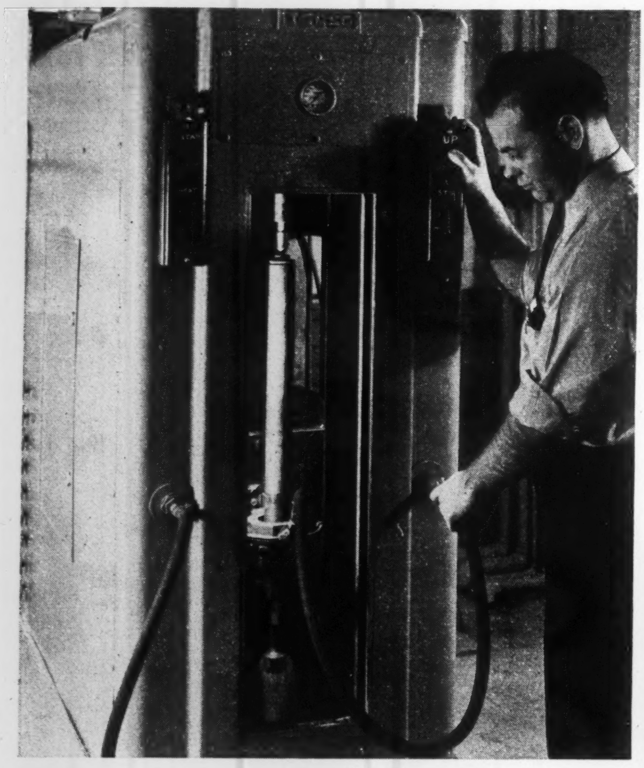


# Applications

## of Engineering Parts, Materials and Process

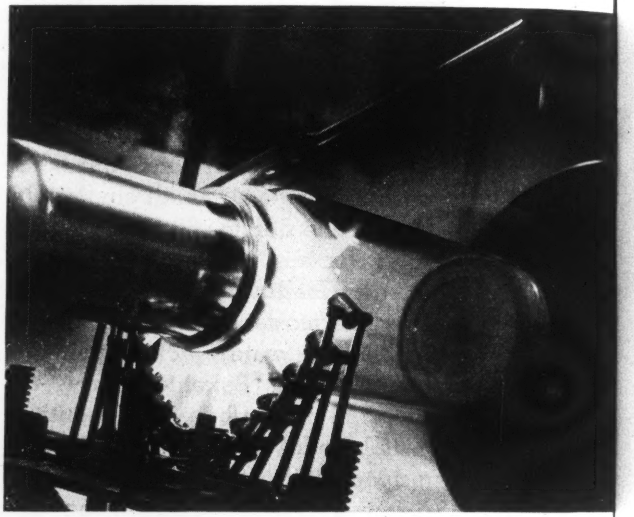
### Metal Bonds to Glass

WITH a coefficient of expansion which closely parallels that of hard glass up to 465 degrees Cent. Kovar alloy of iron, nickel and cobalt makes possible the effective sealing of electronic power tubes. A cleaned glass tube and Kovar metal are heated in a gas flame, the oxidized metal worked into the softened glass as illustrated at the right, and the joint allowed to cool. The surface thus prepared provides a base for attaching a metal socket to the glass by conventional soldering or brazing.



### Hardens Long Bars

THIS screw-driven continuous induction-hardening machine, above, is used in the new Tocco Experimental Laboratory for experimental hardening of steel bars having a length of hardened area considerably greater than their diameter. The inductor, quench apparatus and transformer of the machine travel upward during the processing cycle. Pieces as large as 6 inches in diameter can be hardened over a 30-inch length. Either 3000 or 9600-cycle operation is available as occasion demands.



### Permanently Records Vibrations

A PURELY mechanical device, the Westinghouse Vib graph used as shown below records vibrations on spot even though in some remote corner of a plant far fr laboratory facilities. The Vibrograph writes a perman record of vibrations over a range of 600 to 15000 cycles minute and amplitudes from one ten-thousandth to one-s teenth of an inch. A record and timing wave drawn by stylus on transparent plastic tape one inch wide are re under a low-power microscope.

# Selecting Materials and Parts for Hydraulic Systems

*TO PROVIDE a convenient check-list for the basic materials and parts required in oil-hydraulic systems for machines, the accompanying data sheet has been prepared in tabular form. For more detailed information the designer should consult manufacturers' technical literature*

| Item and Function | Specifications |
|---|---|
| **Pipe for Pressure Lines** | **Wall Thickness**<br>Wall thickness may be calculated from the formula<br>$$t=\frac{pD}{2s}$$<br>where $t$ = Thickness, inches<br>$D$ = Outside diameter of tube, inches<br>$p$ = Maximum pressure, psi<br>$s$ = Allowable working stress, psi.<br>**Seamless Steel Tubing**<br>Preferred wherever possible.<br>Allowable working stress $s$ = 7000 to 10,000 psi for low carbon steel tubing.<br>For use with flared or compression fittings, standard O.D. tubing (tubing with outside diameters which are whole numbers or standard fractions of an inch) is used. Several thicknesses are available in each size.<br>For use with screwed fittings, tubing with outside diameters corresponding to standard pipe thread sizes is used.<br>Radius of pipe bends should be at least 2½ to 3 times the inside diameter of the tube.<br>**Standard Pipe**<br>Standard steel seamless or welded pipe can be used for straight lengths in place of tubing. Outside diameters correspond to standard pipe threads. For each outside diameter three thicknesses are available, designated as *standard*, *extra heavy* and *double extra heavy*. Safe working pressure for any size may be worked out from the formula already given.<br>**Seamless Copper Tubing**<br>Satisfactory for small sizes and odd shapes to reduce cost of fabrication if there is no excessive vibration or shock. Should be protected against damage by being located inside housings and against vibration by clamps and fasteners.<br>Allowable working stress $s$ = 3500 to 5000 psi. Maximum working pressure not over 1000 psi.<br>**Flexible Hose**<br>Low-pressure neoprene-lined hose up to 300 psi.<br>Medium-pressure neoprene-line fabric-braided hose up to 800 psi.<br>High-pressure neoprene-lined metal-braided hose for pressures up to 3000 psi.<br>**Fluid Hammer**<br>Sudden pressure rises may greatly exceed the normal pressure |

# ENGINEERING DATA SHEET

| Item and Function | Specifications |
|---|---|
| **Pipe for Pressure Lines** | in systems for machine tools and other types of equipment, and should be taken into account in applying the formula. |
| **Pipes for Drain Lines** | **Standard Welded Black Steel Pipe**<br>Used in straight sections only after being thoroughly cleaned and pickled.<br>**Seamless Thin-Wall Copper Tubing**<br>May be used for small sizes and odd shapes when properly proected against damage or vibration. |
| **Joints in Pressure Lines** | **Screwed**<br>Used in conjunction with screwed fittings or screwed flange connections, for steel pipes and pipe-size tubing with tapered pipe threads. Threads to be cut with sharp tools and to be free of burrs.<br>**Flared**<br>For standard O.D. tubing, steel or copper.<br>**Compression (Ball Sleeve)**<br>For standard O.D. tubing, steel or copper.<br>**Welded**<br>For larger sizes.<br>**Soldered**<br>For copper tubing up to 250 psi pressure. |
| **Joints in Drain Lines** | **Screwed**<br>For steel pipes and tubing up to and including 2½ inches.<br>**Flanged**<br>For sizes above 2½ inches. Screwed cast iron or welded steel flanges.<br>**Welded**<br>For larger sizes (above 2½ inches).<br>**Flared, Compression**<br>For standard O.D. steel tubing.<br>**Flared, Compression, Soldered**<br>For copper tubing. |
| **Fittings for Pressure Lines** | **Screwed Fittings**<br>Standard screwed malleable iron banded fittings for low pressure.<br>300-pound screwed malleable iron banded fittings for medium pressure.<br>Forged steel hydraulic fittings for high pressure.<br>**Flared Tube Fittings**<br>Steel fittings for steel tubing. Forged or extruded brass fittings for copper tubing.<br>**Soldered Fittings**<br>For copper tubing up to 250 psi pressure.<br>**Welded Fittings**<br>For improving pipe runs by avoiding abrupt changes in flow direction. |
| **Fittings for Drain Lines** | **Screwed Fittings**<br>For use with steel pipe.<br>Standard screwed malleable iron banded fittings up to and including 2½ inches.<br>**Flanged Fittings**<br>Standard cast iron flanged fittings in connection with screwed cast iron flanges above 2 inches.<br>**Flare Tube Fittings**<br>For copper tubing.<br>**Soldered Fittings**<br>For copper tubing. |

# ENGINEERING DATA SHEET

| Item and Function | Specifications |
|---|---|
| Fittings for **Drain Lines** | **Welded Fittings**<br>To improve pipe runs and eliminate joints where suitable. |
| Unions for **Pressure Lines** | **Screwed**<br>Malleable iron screwed unions with brass seating ring and ground joint, for low and medium pressures.<br>**Flanged**<br>Malleable iron flanged unions with brass seating ring and self-seating ball joint, for high pressures.<br>**Flared-Tube, Compression**<br>For steel and copper tubing.<br>**Soldered**<br>For copper tubing. |
| Unions for **Drain Lines** | **Screwed**<br>Standard malleable iron screwed unions with brass seating ring up to and including 2½ inches.<br>**Flanged**<br>Cast iron flanged union with gasket above 2½ inches.<br>**Flared-Tube, Compression**<br>For steel and copper tubing.<br>**Soldered**<br>For copper tubing. |
| **Thread** | **Screwed Pipe**<br>American (National) standard taper pipe thread with appropriate sealer.<br>Dryseal American (National) standard taper pipe thread which, having no clearance at crest and root, can render joints pressure tight without sealer. |
| **Thread Seal** | **Pipe Compound**<br>Oil-resisting compound, should cover 2/3 of thread with 1/3 from end of tube remaining free of compound.<br>**Tinning**<br>Tinning of male thread, preferred for high pressures.<br>**Brazing**<br>Screwed joints made up tight with oil lubricant only, then brazed after assembly. |
| Gaskets* | **Fiber**<br>Fiber, 1/16-inch thick, treated to resist oil, up to and including 250 psi.<br>**Metallic**<br>Copper or aluminum, asbestos millboard filled, above 250 psi. |
| **Bolts for Flanged Connections** | **Steel**<br>American standard, semi-finished, hexagon head bolt and nut. |
| Valves, **Shut-Off** | **Globe and Angle Valves**<br>Angle valves placed where bend naturally occurs add less resistance to flow than globe valve in straight section. |
| Valves, **Selector** | **Plug**<br>Suitable where hydraulic shock must be avoided but subject to leakage as result of wear.<br>**Spool Piston Valve**<br>Low operating force, suitable for actuation by solenoids. Commonly used where leakage unimportant.<br>**Poppet Valve**<br>Suitable for high pressures and thin fluids, will maintain seal for |

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*See the article "Gaskets in Design", M. D., March, 1945, Page 151.

# ENGINEERING DATA SHEET

| Item and Function | Specifications |
|---|---|
| **Valves, Selector** | prolonged period due to absence of wear. |
| **Valves, Metering** | **Needle**<br>For small volume flow. Can shut off without leaking.<br>**Grooved Plunger**<br>For large volume flow at low pressure.<br>**Balanced Pressure Orifice**<br>For precision control, higher pressure.<br>**Choke Coils**<br>Small diameter copper tubing with length to suit desired pressure drop. For limiting flow to one or more units. |
| **Valves, Check** | **Ball Check**<br>For small sizes and low pressures.<br>**Piston or Disk Check**<br>For higher pressures and higher velocities.<br>**Swing Disk**<br>Sometimes preferred because of straight flow, freedom from sticking and low pressure loss. |
| **Valves, Relief** | **Adjustable Spring-Loaded Ball, Spool or Piston Type**<br>Located in pressure line as near as possible to pump. |
| **Valves, Back Pressure** | **Standard Relief Valve**<br>Placed in line for unidirectional flow, set at appropriate relief pressure.<br>**Foot or Sequence Valve**<br>For two-directional control flow, controlled one way only, set at appropriate relief pressure. |
| **Pumps** | **Gear**<br>Ordinarily used for low pressures, up to 300 psi, constant delivery.<br>**Piston**<br>For pressures up to 3000 psi, constant or variable delivery.<br>**Vane**<br>For pressures comparable to piston type. Constant delivery. Especially suited to pumping thin fluids without excessive slip. |
| **Motors** | **Gear**<br>Well adapted to shock and heavy loads, constant displacement.<br>**Piston**<br>Constant or variable displacement.<br>**For Constant Torque Output**<br>Use variable-delivery pump, constant displacement motor.<br>**For Constant Horsepower Output**<br>Use constant-delivery pump, variable-displacement motor. |
| **Cylinders** | **Single-Acting**<br>Unidirectional power stroke, gravity return.<br>**Double-Acting**<br>With differential areas, can provide slow power stroke and quick return. |
| **Filters** | **Reservoir**<br>Provides best protection for pump and system.<br>**Circulating**<br>Placed in system, does not detract from performance of any subsystem. Filters indirectly.<br>**By-Pass**<br>Enables the use of the smallest-size filter when extremely fine filtration necessary and total circulation large. Filters only a small percentage of total flow. |

**MACHINE DESIGN'S**

**MATERIALS WORK SHEET**

FILING NUMBER
**9.02**

# Aluminum-Base Casting Alloys

### ASTM Nos. B26-44T, B108-44T, B85-44T

## AVAILABLE IN:

| | | |
|---|---|---|
| **B26-44T** | (all alloys) | Sand castings |
| **B108-44T** | (all alloys) | Permanent-mold castings |
| **B85-44T** | (all alloys) | Die castings |

## ANALYSES:
See last page of Work Sheet.

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## PROPERTIES

| | | | | Typical Values | | | |
|---|---|---|---|---|---|---|---|
| **SAND CASTINGS ASTM No. B26-44T** Alloys | **Tensile Strength (min, psi)** | **Yield Strength (typical, .2 per cent offset, psi)** | **Elongation in 2 Inches (min, per cent)** | **Brinell Hdns. (10-mm ball, 500-kg load)** | **Shear[1] Strength (psi)** | **Endurance[2] Limit (psi)** | **Charpy Impact (ft-lbs)** |
| C1, Heat treatment No. 1 | 29,000 | 16,000 | 6 | 60 | 24 000 | 6 000 | 2.8 |
| C1, Heat treatment No. 2 | 32,000 | 22,000 | 3 | 75 | 30,000 | 6,500 | 1.8 |
| C1, Heat treatment No. 3 | 36 000 | 31,000 | † | 95 | 31,000 | 7,000 | 1.3 |
| C1, Heat treatment No. 4 | 29 000 | 19,000 | 3 | .. | ...... | ..... | ... |
| C2, Heat treatment No. 1 | 27,000 | 16,000 | 3 | .. | ...... | ..... | ... |
| G1, As-cast | 22 000 | 12,000 | 6 | 50 | 20 000 | 7,000 | 3.8 |
| S1 and S2, As-cast | 17,000 | 9,000 | 3 | 40 | 14,000 | 6,500 | 1 |
| CG1, Heat treatment No. 1 | 23 000 | 20,000 | † | 80 | 21,000 | 9,500 | ... |
| CG1, Heat treatment No. 2 | 30,000 | 30,000 | † | 115 | 29,000 | 8,500 | ... |
| CN21, Heat treatment No. 1 | 23,000 | 20,000 | † | 70 | 21,000 | 6 500 | .. |
| CN21, Heat treatment No. 2 | 32,000 | 32,000 | † | 100 | 32,000 | 8,000 | .6 |
| CS22, As-cast | 19 000 | 14,000 | † | 70 | 20,000 | 8,000 | .6 |
| SC21, Heat treatment No. 1 | 32,000 | 25,000 | 2 | 80 | 30 000 | 8,500 | 1 |
| SC21, Heat treatment No. 2 | 25 000 | 23,000 | † | 65 | 22,000 | 7,000 | .7 |
| SC21, Heat treatment No. 3 | 30,000 | 29,000 | † | 75 | ... | 10,000 | ... |
| SG1, Heat treatment No. 1 | 30 000 | 22,000 | 3 | 70 | 27 000 | 8,000 | 1 |
| SG1, Heat treatment No. 2 | 23,000 | 20,000 | † | 60 | 18,000 | 7,500 | .7 |
| ZG41, Aged | 30,000 | 25,000 | 3 | .. | ...... | ..... | ... |
| **PERMANENT-MOLD CASTINGS ASTM No. B108-44T** | | | | | | | |
| C3, As-cast | 23,000 | 23,000 | † | 80 | 22 000 | 9,500 | ... |
| S1, As-cast | 21,000 | 9,000 | 2.5 | 45 | 18,000 | ..... | 2.1 |
| S2, As-cast | 21,000 | ...... | 2 | 45 | 18,000 | ..... | ... |
| CG1, Heat treatment No. 1 | 30,000 | 30,000 | † | 100 | 25,000 | . | ... |
| CG1, Heat treatment No. 2 | 40,000 | 36,000 | † | 140 | 30 000 | 9,000 | ... |
| CN21, Heat treatment No. 1 | 34,000 | 34,000 | † | 105 | 26,000 | 10,500 | .6 |
| CN21, Heat treatment No. 2 | 40 000 | 42,000 | † | 110 | 31 000 | 9,500 | .6 |
| CS4, Heat treatment No. 1 | 33,000 | 22,000 | 4.5 | 75 | 30 000 | 9,500 | ... |
| CS4, Heat treatment No. 2 | 35,000 | 33,000 | 2 | 90 | 32,000 | 10,000 | ... |
| CS4, Heat treatment No. 3 | 33,000 | ..... | 3 | .. | .... | ..... | ... |
| SC1, As-cast | 23,000 | 16,000 | † | 70 | 25,000 | ..... | .9 |
| SC41, Heat treatment No. 1 | 32,000 | ...... | † | .. | ...... | ..... | ... |
| SC41, Heat treatment No. 2 | 36,000 | ...... | † | .. | ...... | ..... | ... |
| SG1, Heat treatment No. 1 | 33,000 | 27,000 | 3 | 90 | ...... | ..... | 1 |
| SN41, Heat treatment No. 1 | 31,000 | 28,000 | † | 105 | 24,000 | ..... | .7 |
| SN41, Heat treatment No. 2 | 40,000 | ...... | † | .. | ...... | ..... | ... |

| | Typical Values | | | | |
|---|---|---|---|---|---|
| **DIE CASTINGS ASTM No. B85-44T** Alloys | **Tensile[3] Strength (psi)** | **Yield Strength[3] (.2 per cent offset, psi)** | **Elongation[3] in 2 Inches (per cent)** | **Endurance[2] Limit (psi)** | **Charpy[4] Impact (ft-lbs)** |
| S4 | 30 000 | 13,000 | 4.5 | ..... | 4.5 |
| S5 | 37,000 | 18,000 | 1.8 | 15,000 | 2 |
| S9 | 35,000 | ...... | 3.5 | ...... | .. |
| SC2 | 38,000 | ...... | 2.5 | ..... | 2.5 |
| SC5 | 35 000 | 19,000 | 3 | 17,000 | 3 |
| SC6 | 41,000 | ...... | 3 | ...... | 3.5 |
| SC7 | 43,000 | ...... | 2 | ...... | 3.2 |

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[1] These are single-shear values obtained from double-shear tests.

[2] Based on specimens withstanding 500,000,000 cycles of completely reversed stress, using rotating beam type of machine and specimen.

[3] Tests on round specimens.

[4] Tests on notched specimens (10 x 10 cm).

†Not required.

MACHINE DESIGN is pleased to acknowledge the collaboration of Aluminum Company of America in this presentation.

## CONSTANTS

| Alloys | | Specific Gravity | Weight (lb/in³) | Solidification Range (deg F) | Electrical Conductivity (% Int'l Annl'd Cu Std at 68 F) | Thermal Conductivity (C.G.S. units⁵ at 68 F) | Coef. of Thermal Exp. (per deg F x 10⁻⁶) 68-212 F | Coef. of Thermal Exp. (per deg F x 10⁻⁶) 68-572 F |
|---|---|---|---|---|---|---|---|---|
| **SAND CASTINGS, ASTM No. B-26-44T** | | | | | | | | |
| C1, | Heat treatment No. 1 | 2.77 | .1 | 1195-1020 | 35 | .33 | 12.7 | 13 8 |
| | Heat treatment No. 2 | 2.77 | .1 | 1195-1020 | | | 12 7 | 13.8 |
| | Heat treatment No. 3 | 2.77 | .1 | 1195-1020 | 37 | .34 | 12.7 | 13.8 |
| G1, | As-cast | 2.63 | .095 | 1185-1075 | 35 | .32 | 13.3 | 14.4 |
| S1, | As-cast | 2.66 | .096 | 1165-1070 | 37 | .34 | 12.2 | 13.3 |
| CG1, | Heat treatment No. 1 | 2.85 | .103 | 1160-1005 | 41 | .38 | 12.2 | 13 |
| CN21, | Heat treatment No. 1 | 2.73 | .099 | 1165-995 | 44 | .4 | 12.5 | 13.6 |
| | Heat treatment No. 2 | 2.73 | .099 | 1165-995 | 37 | .35 | 12.5 | 13.6 |
| CS22, | As-cast | 2.85 | .103 | 1175-1005 | 30 | .28 | 12.2 | 13.3 |
| SC21. | Heat treatment No. 1 | 2.68 | .097 | 1160-1075 | 36 | .33 | 12.2 | 13.3 |
| | Heat treatment No. 2 | 2.68 | .097 | 1160-1075 | 43 | .4 | 12.2 | 13.3 |
| SG1, | Heat treatment No. 1 | 2.63 | .095 | 1130-1075 | 39 | .36 | 11.9 | 13 |
| | Heat treatment No. 2 | 2.63 | .095 | 1130-1075 | 43 | .39 | 11.9 | 13 |
| **PERMANENT-MOLD CASTINGS, ASTM No. B108-44T** | | | | | | | | |
| C3, | As-cast | 2.86 | .103 | 1165-975 | 27 | .26 | 12.2 | 13.3 |
| S1, | As-cast | 2.68 | .097 | 1165-1070 | 41 | .38 | 12.2 | 13.3 |
| CG1, | As-cast | 2.89 | .104 | 1160-1005 | 34 | .32 | 12.2 | 13 |
| CN21, | Heat treatment No. 1<br>Heat treatment No. 2 | 2.77 | .1 | 1165-995 | 34 | .32 | 12.5 | 13 |
| CS4, | Heat treatment No. 1 | 2.78 | .101 | 1160-980 | 35 | .33 | 12.2 | 13.3 |
| | Heat treatment No. 2 | 2.78 | .101 | 1160-980 | 50 | .45 | 12.2 | 13.3 |
| SC1, | As-cast | 2.77 | .1 | ........ | 37 | .34 | 11.9 | 12.7 |
| SG1, | Heat treatment No. 1 | 2.63 | .095 | 1130-1075 | 41 | .38 | 11.9 | 13 |
| SN41, | Heat treatment No. 1 | 2.68 | .097 | 1095-1000 | 29 | .28 | 10.5 | 11.6 |
| **DIE CASTINGS, ASTM No. B85-44T** | | | | | | | | |
| S4 | | 2.7 | .097 | 1165-1070 | 41 | .38 | 12.2 | 13.3 |
| S5 | | 2.66 | .096 | 1150-1070 | 36 | .33 | 11.1 | 11.9 |
| S9 | | ... | ... | ........ | .. | .. | ... | ... |
| SC2 | | 2.78 | .101 | ........ | 28 | .27 | 11.6 | 12.7 |
| SC5 | | ... | ... | ........ | .. | .. | ... | ... |
| SC6 | | ... | ... | ........ | .. | .. | ... | ... |
| SC7 | | ... | ... | ........ | .. | .. | ... | ... |

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[5] C.G.S. units are calories per square centimeter per degree centigrade per second.

## CHARACTERISTICS

Aluminum-base casting alloys combine good mechanical properties with excellent casting and machining characteristics. They are of two types: Common alloys with properties resulting solely from alloying additions, and heat-treatable alloys, in which the high strength is largely the result of solution heat treatment and natural or artificial aging. While the corrosion resistance of these alloys is good, protective coatings sometimes are desirable under severe service conditions (see Corrosion Resistance). Alloys covered in this Work Sheet are for sand, permanent-mold, and die casting. In general, the mechanical properties of castings made in metal molds are superior to those of the same alloy cast in sand.

**Aluminum-Copper Alloys** (C1, C2, CG1, CS22, C3): Most widely used of all the casting alloys. Types C1 and C2 are used to produce sand castings in several heat-treated conditions. These alloys have good casting and machining characteristics. Alloy CG1 provides the maximum in hardness and wear resistance of this group as well as good streng at elevated temperatures in both sand and permanent-mo castings. Alloy C3 is especially suited to permanent-mo casting. Aluminum-copper alloys are not as resistant corrosion as are the aluminum-silicon or aluminum-ma nesium types, nor do they take anodic coatings that a as thick and abrasion-resistant.

**Aluminum-Silicon Alloys** (S5, S9, S1, S2, S4, SG1, SN4 The excellent casting characteristics of these alloys ma them especially suitable for intricate sand, permanent-mo and die castings. Leakproof castings can be produced a are used widely for pressure and vacuum fittings. Alu inum-silicon alloys show excellent resistance to cor sion, hence are well suited for marine parts, e Under severely corrosive conditions protective fi ishes are used. Alloys in this class can be m chined satisfactorily though not as readily as the oth types. Tensile properties are somewhat lower than tho of the aluminum-copper group, although Alloy SG1 in t heat treated condition has comparable properties. All

41 has the lowest coefficient of thermal expansion of of the commercial alloys, possesses excellent wearing lities and retains its strength well at elevated temratures.

**luminum-Magnesium Alloy** (G1): Considerably superior all the other common sand casting types in its resistce to corrosion and tarnish. However, its casting propies are not as good as many of the other alloys and re exacting melting and foundry practices are required consistent results. Rapid chilling and more thorough ding are necessary to obtain maximum soundness with se castings than with the other alloys. Thus, leakproof ts are difficult to make. This alloy responds well to face treatment processes.

**luminum-Copper-Nickel-Magnesium Alloy** (CN21): Most quently used for castings that must retain their strength elevated temperatures and where good bearing characistics are required. Foundry characteristics are similar those of the aluminum-magnesium alloys. It is not table for leakproof castings, particularly those with thin ll sections.

**luminum-Copper-Silicon Alloys** (CS4, SC1, SC2, SC5, 6, SC7, SC21, SC41): These alloys have better cast characteristics than the aluminum-copper group. Meanical properties and machining qualities are superior those of the aluminum-silicon alloys. Corrosion resistce generally is better than that of aluminum-copper als but somewhat inferior to that of aluminum-silicon oys. These alloys produce sound pressure-tight castings intricate design.

**luminum-Zinc Alloy** (ZG41): A casting alloy providing h strength approaching that of many of the heat treated oys, without heat treatment. Material ages rapidly at m temperature, substantially reducing ductility. It has tter resistance to corrosion than aluminum-copper alloys t inferior to aluminum-silicon and aluminum-magnesium oys.

## APPLICATIONS

**lloys C1, C2:** Used for heat-treated sand castings where od shock resistance and corrosion resistance are reired. Not suitable for permanent-mold or die casting.

**lloy CG1:** Suited for both sand and permanent-mold stings where high strength is necessary at elevated temratures or where good hardness and wear resistance are quired. Permanent-mold castings include automotive pisns. Sand castings include air-cooled cylinder heads and stons.

**lloy CS22:** For sand-cast parts where high impact is not countered. Used for automotive crankcases, oil pans, anifolds, transmission housings, miscellaneous generalrpose machine castings.

**lloy C3:** For permanent-mold cast parts where high pact is not anticipated. Used for automotive cylinder ads, washing machine agitators, vacuum cleaner housgs, and general-purpose castings with pressure requireents.

**lloys S5, S9:** For general-purpose die castings that are ge and intricate in design and where high resistance to rosion is necessary.

**lloys S1, S2:** Suited for both sand and permanent-mold stings where good ductility and shock resistance are reired. Have excellent corrosion resistance, thus are used r sand-cast pipe fittings, food handling equipment and marine fittings. Permanent-mold castings include general-purpose parts having thin sections.

**Alloy S4:** For general-purpose die castings such as household equipment parts, automobile parts, brackets, etc. Not used as widely as S1, SC2 and SC7.

**Alloy SG1:** Used for sand and permanent-mold castings much the same as are alloys C1 and C2, especially where resistance to severe corrosion is required and intricacy of design demands its superior casting characteristics.

**Alloy SN41:** For permanent-mold castings where low expansion and strength at elevated temperatures are required, as in automotive pistons, etc.

**Alloy G1:** Employed for sand castings where good corrosion resistance is necessary but pressure-tightness is not a factor. Hence it is well suited for castings for sewage-disposal plants, chemical plants, dairy equipment, and marine parts.

**Alloy CN21:** For both sand and permanent-mold castings where strength at elevated temperatures is necessary. Sand castings include pistons and air-cooled cylinder heads. Permanent-mold castings include pistons for both diesel and gasoline engines.

**Alloy SC21:** For sand and permanent-mold castings. Uses, similar to those for alloy SG1, also include crankcases, cylinder heads and similar parts for liquid-cooled aircraft engines.

**Alloy SC41:** For permanent-mold-cast parts of intricate design requiring pressure tightness and moderate corrosion resistance.

**Alloy CS4:** A modified version of alloys C1 and C2 developed for permanent-mold work. Castings of this alloy, after solution heat treatment, have appreciably higher tensile properties than those of similarly treated C1 and C2.

**Alloy SC1:** For intricate permanent-mold castings such as general-purpose parts.

**Alloys SC6, SC7:** General-purpose die-casting alloys used where better castability is required than is obtained with SC2 and resistance to corrosion somewhat less than obtained with S5 is satisfactory.

**Alloy ZG41:** Used for general-purpose sand castings.

**Alloy SC2, SC5:** For die-cast brackets, frames and levers with thick sections.

## FABRICATION

### SAND CASTING:

Foundry equipment for casting aluminum-base alloys is not unlike that for casting other metals. Casting shrinkage for aluminum-base alloys usually is figured at 5/32-inch per linear foot for average castings made in green sand. Total contraction will depend on size, shape and method of molding.

From the economy standpoint, green-sand molds and cores are preferred. Inserts of other metals can be cast into aluminum-base alloy castings, cast iron and steel inserts being the most suitable, while brass and bronze inserts should be avoided. Inserts should be knurled or grooved to insure good mechanical bond. Although sections as thin as ⅛-inch have been cast, 3/16-inch is considered the practical minimum for small and medium-size castings. Drastic variations in section thickness result in casting difficulties.

(*Continued on next page*)

## PERMANENT-MOLD CASTING:

Wherever it can be justified, the permanent-mold or semipermanent-mold process should be used. It offers not only the advantages of superior metallurgical and mechanical properties but also closer dimensional tolerances. Finish-machining costs on such parts usually are materially lower than those for sandcast parts. Shrinkage in permanent molding must be determined by experience.

## DIE CASTING:

Smoother surface finish with permanence and accuracy of dimensions, in addition to great uniformity at an extremely rapid production rate, are characteristics which suit the die-casting process admirably for large quantity production. Machining can be reduced or totally eliminated. Threads and cored holes can be cast as well as studs and inserts of dissimilar metals. In addition, extremely thin sections can be cast. On large parts sections as thin as .08-inch may be cast and, where sections are less than six inches in length, the thickness may be as low as .05-inch. Uniform wall thickness is desirable throughout.

Die castings up to 20 pounds in weight have been produced and some with a length of over 40 inches. Commercial castings generally are produced with tolerance limits of plus or minus .0015-inch per linear inch of dimension parallel to the parting line. Dimensions perpendicular to the parting line are held to minus zero, plus .01-inch in large parts and minus zero to plus .003-inch in small ones. Shrinkage effects must be determined from experience.

## MACHINABILITY:

All of the aluminum-base casting alloys have excellent machining characteristics except those in the aluminum-silicon group. However, the alloys in this group can be machined satisfactorily through the use of proper techniques, tools and lubricants. The ease with which most of these alloys are finished is one of their important economies and their light weight facilitates handling during machining. Smoothness of finish, absence of mold-sand particles and the nonabrasive character of the metal makes for economical tooling.

Tungsten-carbide tools offer advantages, especially where good finish is required on aluminum-silicon alloys. Tools should have more side and top rake than those for steel. Edges should be keen and smooth with stoned edges. Best results are obtained with high speeds and fine to medium feeds. Copious quantities of lubricant should be used with the work held rigidly to avoid vibration.

## WELDING:

Gas or electric-arc welding can be applied to the joining of castings or assemblies using castings. For gas welding the oxyacetylene flame is preferred. Both metallic and carbon arcs are well suited to various casting alloys. The carbon arc produces welds more nearly free of oxides and porosity. Metallic arc is faster and more flexible. Preheating temperatures up to 700 to 800 degrees Fahr. usually are required to eliminate thermal stresses with either method.

Castings to be welded should be carefully designed to allow easy access for welding. Wherever possible, welds should be made in sections allowing cooling strains to be taken by the casting rather than the weld itself. Parts should be designed so that the surfaces to be joined are as nearly the same thickness as possible. Where castings assembled by welding require a subsequent heat treatm a welding rod of the parent alloy should be used. general, 3/16 to 1/4-inch rods are satisfactory for average-size weldments.

## BRAZING:

Brazing of aluminum-base castings is still in the exp mental stage and as yet cannot be called a generally process. Melting ranges of most of the casting alloys quire a brazing wire with a melting point considera below that of the materials developed to date. Alloys S2 and ZG41 have been successfully brazed, but the cas producer should be consulted before attempting any b ing job.

## RESISTANCE TO CORROSION

Commercially pure aluminum and also all of its cast alloys are resistant to atmospheric corrosion to a mar degree. This is particularly true of the aluminum-sili alloys and to a greater extent to the aluminum-magnesi alloys. Because of this excellent resistance to corros attack, alloys S1, S2, S4, SG1 and G1 are preferred to other alloys in producing castings subject to outdoor posure. This applies for example, to marine castings wh attack by salt-laden atmosphere is most likely to occ

For most applications a treatment of dilute aque solution of phosphoric acid gives adequate surface p paration for subsequent protective paint coatings. Ho ever, for extremely severe service conditions where pa are constantly wet or subject to high humidity, anodic chemical oxide coating is advantageous in promoting hesion of paint and protecting the alloy.

## GALVANIC CORROSION

Aluminum-base alloys are anodic to most other co monly used metals and alloys (except magnesium, zinc, c mium, and their alloys). This means that, when expos in the presence of moisture or conducting liquids, asse blies made up of aluminum alloys and other metals especially copper or copper-base alloys—may suffer fr galvanic corrosion. Such galvanic corrosion may be p ticularly serious in seacoast or sea water environme For this reason, aluminum alloy parts never should used in contact with parts made of copper or brass marine exposures unless suitable protective measures employed.

Appreciable galvanic action on the various alumin alloys may also result by contact with steel, nickel, tin, lead, although generally this is less severe than that caus by contact with copper-base alloys. Contact with ca mium or cadmium-plated parts results in little or no g vanic attack of aluminum alloys. Zinc is anodic to alu inum-base alloys under most exposure conditions. The fore, contact with zinc (or zinc alloys) generally results accelerated corrosion of the zinc-base materials and p tection of the aluminum-base alloys. Magnesium and alloys are also anodic to aluminum alloys. However, co tact between magnesium alloys and aluminum alloys m result in accelerated corrosion of both alloys unless su able protective measures are used. The aluminum-ba alloys containing magnesium generally cause less galva action when in contact with magnesium-base alloys th do the other aluminum-base alloys.

# ANALYSES

## SAND CASTINGS

| ASTM No. B26-44T | | Cu | Fe | Si | Mg | Mn | Zn | Ni | Ti | Al | Others** (total) |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Alloy | C1 .... | 4-5 | 1 | 1.5 | .03 | .3 | .3 | .... | .2 | * | .15 |
| | C2 .... | 3-4.5 | 1.2 | 2.5 | .05 | .5 | .5 | .3 | .2 | * | .3 |
| | G1 .... | .1 | .5 | .3 | 3.5-4 5 | .6 | .1 | ..... | .2 | * | .15 |
| | S1 .... | .1 | .8 | 4.5- 6 | .05 | .3 | .3 | ..... | .2 | * | .. |
| | S2 .... | .4 | .8 | 4.5- 6 | .05 | .3 | .3 | .... | .2 | * | .3 |
| | CG1 .... | 9.2-10 8 | 1.5 | 1 | .15-.35 | .3 | .4 | .3 | .2 | * | .3 |
| | CN21 .... | 3.5- 4.5 | 1 | .7 | 1.2-1.8 | .3 | .1 | 1.7-2.3 | .2 | * | .15† |
| | CS22 .... | 6-8 | 1.4 | 1-3 | .07 | .5 | 2.5 | .3 | .2 | * | .5 |
| | SC21 .... | 1- 1.5 | .6 | 4.5-5.5 | .4- .6 | .1 | .3 | ..... | .2 | * | .15† |
| | SG1 .... | .2 | .6 | 6.5-7.5 | .2- .4 | .3 | .3 | ..... | .2 | * | .15 |
| | ZG41 .... | .2 | 1 | .25 | .5- .7 | ... | 5-6 | ..... | .1-.3 | * | ‡ |

## PERMANENT-MOLD CASTINGS

| ASTM No. B108-44T | | Cu | Fe | Si | Mg | Mn | Zn | Ni | Ti | Al | Others** (total) |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Alloy | C3 .... | 6-8 | 1.4 | 4 | .07 | .5 | 2.5 | .3 | .2 | * | .5 |
| | S1 .... | .1 | .8 | 4.5- 6 | .05 | .3 | .3 | ..... | .2 | * | .. |
| | S2 .... | .4 | .8 | 4.5- 6 | .05 | .3 | .3 | ..... | .2 | * | .3 |
| | CG1 .... | 9.2-10.8 | 1.5 | 1 | .15- 35 | .3 | .4 | .3 | .2 | * | .3 |
| | CN21 .... | 3.5- 4.5 | 1 | .7 | 1.2-1.8 | .3 | .1 | 1.7-2.3 | .2 | * | .15† |
| | CS4 .... | 4-5 | 1.2 | 2-3 | .05 | 3 | .3 | .3 | .2 | * | .3 |
| | SC1 .... | 4-5 | 1 | 5-6 | .1 | .3 | .5 | ... | .2 | * | .5 |
| | SC41 .... | 1-2 | .9 | 11.5- 13 | .4- .7 | .6-.9 | .4 | .05 | .2 | * | .5 |
| | SG1 .... | .2 | .6 | 6.5-7.5 | .2- .4 | .3 | .3 | ..... | .2 | * | .15 |
| | SN41 .... | .5- 1.5 | 1.3 | 11- 13 | .7-1.3 | .1 | .1 | 2-3 | .2 | * | .. |

## DIE CASTINGS

| ASTM No. B85-44T | | Cu | Fe | Si | Mg | Mn | Zn | Ni | Sn | Al | Others** (total) |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Alloy | S4 .... | .6 | 2 | 4.5- 6 | .1 | .3 | .5 | .5 | .1 | * | .2 |
| | S5 .... | .6 | 2 | 11- 13 | .1 | .3 | .5 | .5 | .1 | * | .2 |
| | S9 .... | .6 | 1.3 | 11- 13 | .1 | .3 | .5 | .5 | .1 | * | .2 |
| | SC2 .... | 3-4 | 2 | 4.5-5.5 | .1 | .5 | 1 | .5 | .3 | * | .5 |
| | SC5 .... | 3-4 | 1.3 | 4.5-5.5 | .1 | .5 | .6 | .5 | .3 | * | .5 |
| | SC6 .... | 3-4 | 1.3 | 7.5-9.5 | .1 | .5 | .6 | .5 | .3 | * | .5 |
| | SC7 .... | 3-4 | 2 | 7.5-9.5 | .1 | .5 | 1 | .5 | .3 | * | .5 |

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Note: Where single units are shown, they indicate maximum amounts permitted.

* Aluminum constitutes remainder in all alloys.

** Any one of these elements not to be present in excess of .05 per cent.

† Chromium .2 per cent.

‡ Chromium .4 to .6 per cent.

# MATERIAL DESIGNATIONS

| TM<br>o. | Army-Navy Aero. Board | U. S. Air Forces | Federal | Alum. Co. of America | Aero. Mat'l. Spec's. (SAE) | U. S. Navy | SAE |
|---|---|---|---|---|---|---|---|
| ND CASTINGS, B26-44T | | | | | | | |
| Alloy C1....... | AN-QQ-A-390 | ......... | QQ-A-601,Cl.4 | 195 | AMS-4231 | 46Al(INT)Cl.4 | 38 |
| Alloy C2....... | AN-A-5 | ......... | E-QQ-A-601,Cl.4 | 105 | AMS-4234 | 46Al(INT)Cl.4A | 325 |
| Alloy G1....... | AN-A-26 | ......... | QQ-A-601,Cl.5 | 214 | ........ | 46Ale Cl 5<br>46Al(INT)Cl.5 | 320 |
| Alloy S1....... | AN-A-27 | ......... | QQ-A-601,Cl.2 | 43 | ........ | 46Ale,Cl.2 | 35 |
| Alloy S2....... | .............. | ......... | ............ | ... | ........ | 46Al(INT)Cl.2 | |
| Alloy CG1....... | .............. | ......... | QQ-A-601,Cl.7 | 122 | ........ | ............. | 34 |
| Alloy CN21...... | AN-QQ-A-379 | ......... | QQ-A-601,Cl.6 | 142 | AMS-4220A | ............. | 39 |
| Alloy CS22...... | AN-A-28 | ......... | ........... | 113 | ........ | ............. | 33 |
| Alloy SC21...... | AN-QQ-A-376 | ......... | QQ-A-601,Cl.10 | 355 | AMS-4210B<br>AMS-4212B<br>AMS-4214A | ............. | 322 |
| Alloy SG1....... | AN-QQ-A-394 | ......... | QQ-A-601,Cl.3 | 356 | AMS-4217 | 46Ale Cl.3<br>46Al(INT)Cl.3 | 323 |
| Alloy ZG41...... | AN-A-17 | ......... | ............ | ... | ........ | 46Al(INT)Cl.1 | 310 |
| RMANENT-MOLD CASTINGS, B108-44T | | | | | | | |
| Alloy C3....... | .............. | ......... | QQ-A-596,Cl.1 | C113 | ........ | 46Al5,Cl.1<br>46Al5(INT)Cl.1 | ... |
| Alloy S1....... | .............. | ......... | QQ-A-596,Cl.7 | 43 | ........ | 46Al5 Cl.7 | 35 |
| Alloy S2....... | .............. | ......... | ............ | ... | ........ | 46Al5(INT)Cl.7 | ... |
| Alloy CG1....... | .............. | ......... | QQ-A-596,Cl.2 | 122 | ........ | 46Al5,Cl.2<br>46Al5(INT)Cl.2 | 34 |
| Alloy CN21...... | AN-QQ-A-379 | ......... | QQ-A-596,Cl.3 | 142 | ........ | 46Al5,Cl.3<br>46Al5(INT)Cl.3 | 39 |
| Alloy CS4...... | AN-QQ-A-383 | ......... | QQ-A-596,Cl.4 | B195 | AMS-4282A | 46Al5,Cl.4<br>46Al5(INT)Cl.4<br>OS-839 | 380 |
| Alloy SC1....... | .............. | ......... | QQ-A-596,Cl.5 | A108 | ........ | 46Al5,Cl.5<br>46Al5(INT)Cl.5 | ... |
| Alloy SC41...... | .............. | ......... | QQ-A-596,Cl.10 | ... | ........ | 46Al5,Cl.10<br>46Al5(INT)Cl.10 | ... |
| Alloy SG1....... | .............. | ......... | QQ-A-596,Cl.8 | 356 | AMS-4284 | 46Al5,Cl.8<br>46Al5(INT)Cl.8 | 323 |
| Alloy SN41...... | AN-A-25 | ......... | QQ-A-596,Cl.9 | A132 | ........ | 46Al5,Cl 9<br>46Al5(INT)Cl.9 | 321 |
| E CASTINGS, B85-44T | | | | | | | |
| Alloy S4....... | .............. | 11347(43) | QQ-A-591,Cl.8 | 43 | ........ | 46Al4,Cl.3<br>46Al4(INT)Cl.3 | 304 |
| Alloy S5....... | AN-QQ-A-366(13) | ......... | QQ-A-591,Cl.1 | 13 | AMS-4290A | 46Al4(INT)Cl.1 | 305 |
| Alloy S9....... | AN-QQ-A-366(13X) | 11347(13X) | QQ-A-591,Cl.2 | A13 | AMS-4292 | 46Al4(INT)Cl.2 | .. |
| Alloy SC2....... | AN-QQ-A-366(85) | ......... | QQ-A-591,Cl.5 | 85 | ........ | 46Al4,Cl.5<br>46Al4(INT)Cl.5 | 307 |
| Alloy SC6....... | AN-QQ-A-366(85X) | ......... | QQ-A-591,Cl.10 | ... | AMS-4291 | ............. | E-306 |
| Alloy SC7....... | .............. | ......... | QQ-A-591,Cl.11 | ... | ........ | ............. | E-308 |

# PROFESSIONAL VIEWPOINTS

## ". . . pursuing a new course"

*To the Editor:*

We find the article "Effect of Vibration on Brackets, Fastenings" by A. M. Wahl, outlining the steps desirable to locate and correct causes of fatigue failures in generator supporting members, interesting and informative. The conventional generator bracket, which is essentially a double-flanged spool, is at best an awkward structural member whose size is predetermined by existing engine pad and generator envelope standards. While generators have become longer and heavier with increased output, the bracket has had no opportunity to grow and consequently has become the most critical structural component of the assembly. As usual, engineering finds itself deviled by the choice between making the best of a bad situation or of pursuing some new course. Mr. Wahl is to be congratulated for his thorough and skillful example of the former approach.

As an illustration of the latter course, our engineers felt, when preparing to enter the aircraft generator field, that any alternative might be preferable to the conventional generator bracket. We place little emphasis upon the acceleration, per se, to be experienced by an accessory. Primary design consideration rests upon accessory natural frequency, on the basis that failures are caused by resonant bending stresses exceeding a fatigue limit. This philosophy has been ably reported by John M. Tyler[*], who points out that dangerous bending amplitudes occur only when an accessory's installed natural frequency happens to lie near a shaking or whirl resonance of the engine-propeller combination.

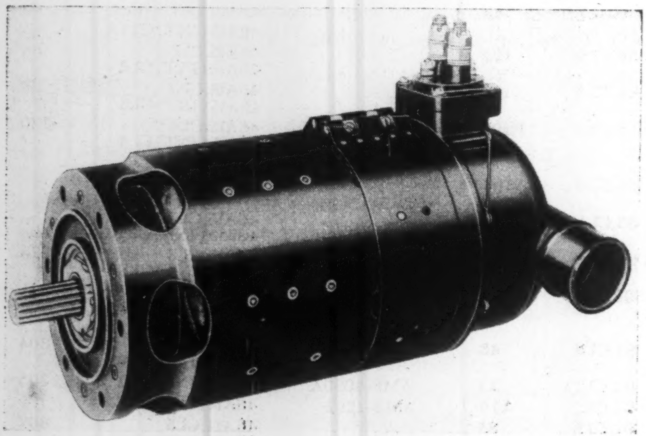

*Fig. 1—Generator design utilizing the highest natural frequency theory which eliminates the spool bracket*

Fortunately for the accessory designer, some ty engines have vibration resonance spectra showing a tude peaks decreasing with frequency. With higher quencies the shaking amplitudes become sufficiently s

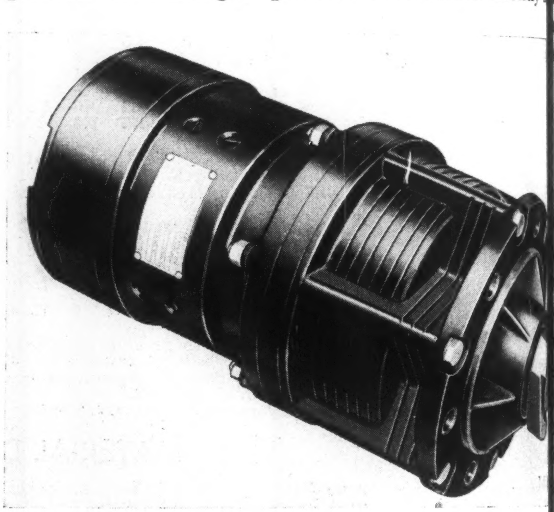

*Fig. 2—Redesign of the starter according to the new th effected a 73 per cent weight saving in the brack*

that bending stresses above the endurance limit are no perienced even with the amplitude magnifications of cessory resonance. Thus the accessory design prob resolves itself into either preparing to withstand amplitudes toward the lower end of the engine frequ spectrum, or of establishing a natural frequency hi than that of any significant engine harmonic. Use of conventional spool-shaped bracket, made of any kn material, on high output generators commits the m facturer to the former of these choices, with the atten difficulties and necessary corrective design refinement parent from Mr. Wahl's paper.

The logical conclusion of this reasoning is a design curing a sufficiently high natural frequency that any ous engine resonance is avoided. Our generator acc plishing this is shown in *Fig.* 1. It will be noticed th gain of about 30 per cent has been possible in the diam of the cylinder section, with of course a much larger in stiffness. Furthermore, the frame and mounting fl are welded together, eliminating both the weakness expense of capscrews.

Aircraft engine starters are another line of our m facture involving similar design problems. The stand mounting bracket or "base" was of forged steel to ob adequate stiffness and endurance strength. However the newest starter, shown in *Fig.* 2, the theory of hig natural frequency has again been applied. The bas is seen to have utilized the maximum available diam-. The six ribs are reinforced internally by steel ugh-bolts and the base material is die-cast magnesium. s starter will crank the same engines as the older model, its installed natural frequencies are on the order of ce as high as those of the older starter. The die-cast gnesium base weighs 1.7 pounds whereas the forged l base weighs 6.3 pounds. For an accessory with a l weight of about 30 pounds this represents a most rthwhile saving.

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[*] "Aircraft Engine Accessory Vibration"—John M. Tyler, Technical Paper No. 44-227, Dec., 1944, supplement to *Transactions of the AIEE.*

hese developments have produced accessories with untings so stiff that much of their installed flexibility within the engine crankcase. Through the use of such essories, the engine rear cover and studs may become structural limit of accessory size. This situation is so irable that the latest specifications for a 30 per cent vier generator stipulate this new (highest natural fre-ncy) frame construction.

—Edwin H. Olmstead, *Project Engineer*
Jack & Heintz, Inc.

*the Editor*:

e cannot wholly agree with Mr. Olmstead's contention t the conventional generator bracket is "....at best an kward structure". In the application of a particular erator to many different engine and airplane combina-ns, it is essential that the engine mounting studs for at-hing the generator be freely accessible. When a me-nic is unduly hampered by inadequate wrench clear-e and extremely cold weather, it may take hours to in-l a generator on an engine. Also, the variety of engine tallations is such that flexibility of mounting position of the electrical connector is essential. Such flexibility is best provided by a large number of mounting holes.

The argument that coincidence between the accessory first critical frequency and any appreciable excitation amplitude can be avoided by raising the natural frequency of the accessory on its mounting (the engine rear cover), seems to us to be misleading. On the basis of information contained in Mr. Tyler's article, it is questionable whether the natural frequency of any large starter or generator on present mounting pads can be raised beyond 15,000 cycles per minute irrespective of the type of mounting flange used. This is due to the flexibility of the engine rear cover and vibration coupling between the different accessories. In engines having nine cylinders in one or two banks, the 5 1/2 order vibration will produce amplitudes of appreciable magnitude at frequencies up to about 15,000 cycles per minute. It would, therefore, be impossible to avoid resonance on these engines.

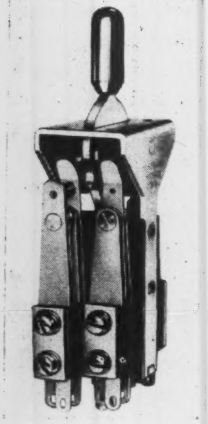
Furthermore, resonance at these higher frequencies will produce alternating loads on the mounting flange and other structural components of the accessory, and on the engine mounting studs which frequently are larger than those produced with an accessory having a lower natural frequency. Such loads are a function of the exciting acceleration imposed on the accessory by the motion of the engine. The exciting acceleration is a function of the amplitude and square of the exciting frequency it may be seen, therefore, whereas the exciting amplitudes are usually lower at the higher frquencies it does not necessarily follow that the exciting accelerations and the resulting alternating forces will also be smaller.

—J. E. Mulheim, *Mechanical Engineer*
*Westinghouse Electric Corp.*

*"The chief told him to dream up a new design"*

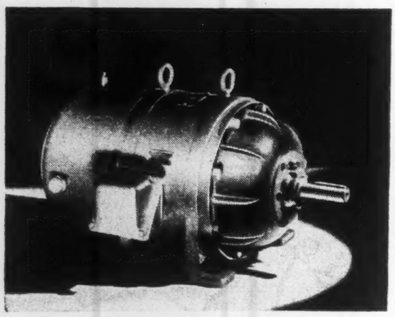
# NEW PARTS AND MATERIAL

## Heavy-Duty Lever Switch



**S**ERIES 4101 extra-heavy-duty lever switch of Donald P. Mossman Inc., 612 North Michigan avenue, Chicago 11, for motor starting, stopping, reversing and motor plugging, affords flexibility of circuit arrangements and almost unlimited combinations of contact assemblies. These may be built up to suit specific needs. The switch shown has contacts rated at 20 amperes, 110 volts alternating current (noninductive). Contacts of 10 amperes are also available.

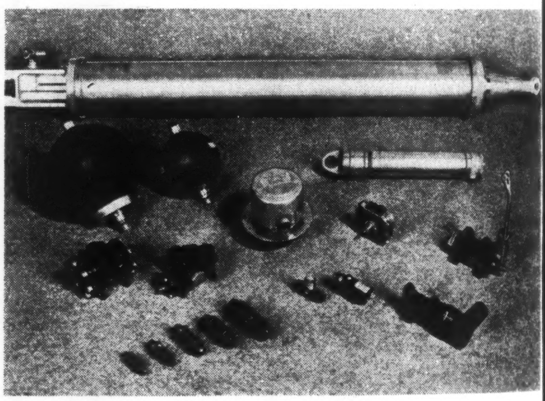
## Explosionproof Motor



**F**OR ATMOSPHERES containing ethyl-ether and similar vapor, the new Class I, Group C, explosionproof motors built by General Electric Co. are furnished in frame sizes 204-326, and in Types K (normal torque) and KG (high starting torque). They are rated at 1/4-horsepower at 600 revolutions per minute to 20 horsepower at 3600 revolutions per minute, 110, 208, 220/440, 550 volts, 60, 50, and 25 cycles. Construction is the same as the Class I, Group D motors, but a special flame path is used at the shaft openings to cool internal explosions. Thermostats are provided to operate in conjunction with a magnetic controller to disconnect the motor at a predetermined temperature, but do not supplant regular overload relays. Included in the motor are other features such as a cast-iron stator, a four-position cast-iron conduit box, and strong end shields of corrosion-resistant cast iron with reinforcing cooling ribs. A nonsparking fan provides cooling, and a pressure-relief system provides for ease of lubrication with disassembly. This motor is recommended for chemical plants, hospi and research laboratories.

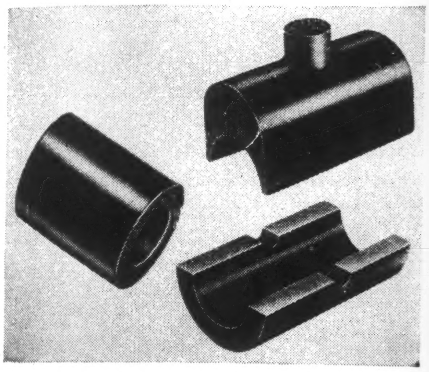
## Hydraulic System Equipment

**T**WENTY UNITS for hydraulic systems up to 30 pounds per square inch pressure are being manufactu by the Pacific Division, Bendix Aviation Corp., No Hollywood, Calif. These include 5, 7 1/2 and 10-inch pr sure accumulators, five sizes of check valves, actuati cylinders, electrically operated 4-way selector valves, fl equalizers, hand pumps, power brake valves, pressu regulators, one-way and two-way restrictor valves, s quence valves, and 3/8 and 1/2-inch 4-way selector valv All the units incorporate the same basic principles of d sign and construction as the company's equipment f 1500 pounds per square inch pressure. Plastic poppe flash-welded cylinder barrels, continuous accumulat shells and flow equalizers are in the new system.



## Self-Lubricating Bearings

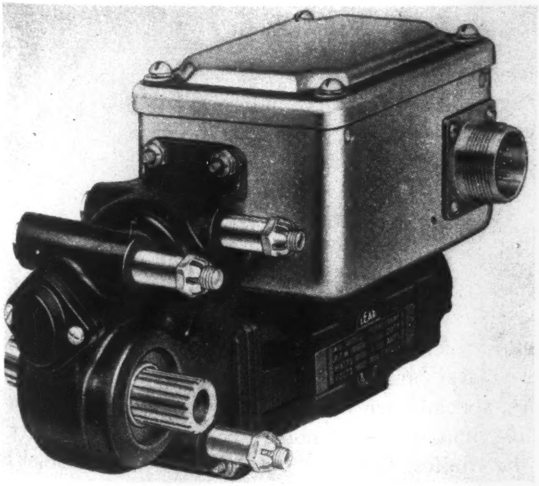
**A**VAILABLE IN compositions of carbon, graphite both, the new bearings of Morganite Brush Co. Inc., Lo Island City, N. Y., are also supplied specially with va ous metallic contents for specific requirements. Some the properties of the bearings are self-lubrication, i munity to acids, alkali and most solvents. They have lo coefficients of friction and expansion. Wear is negligib on the part of the bearings as well as contacting surface Since the bearings impart no odor or taste to liquid they are desirable and widely used in chemical and foo processing industries. They may be machined to clo uracy when desired, or superfinished. Outside diam- rs of ⅛-inch to 15 inches are available. Supplied any shape, complex sleeve or split types, with any con- r required, they may be plated, take molded-in fasten- s and can be shrunk, press fitted, or vulcanized to bber. Operating speeds to 50,000 revolutions per min-



e, with thrust loads to 30 pounds per square inch or ds to 2000 pounds per square inch at lower speeds are ssible.

## Low-Speed Rotary Actuator

DAPTABLE TO bellcrank systems, sprocket chain ives, direct hinge drives for intercooler shutter con- ols, coolant flap controls, landing gears, wing flaps, and her purposes, the new Model 111 series low-speed, tary actuator, now being announced by Lear Inc., qua, O., has been in production for some time. Han- ing loads to 600 pound-inches, it operates at 2.2 revolu- ns per minute. Speed and output torque is varied by ing various gear ratios. Models are equipped with a



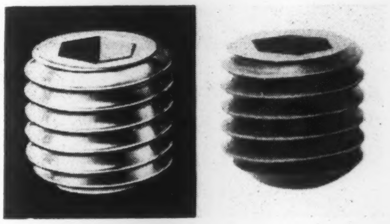
uble-ended ⅝-inch outside diameter S.A.E. 16-tooth line, but special output shafts are available. Output aft rotation normally is less than 360 degrees, th reversible action. The C frame motors may designed with as much as 30 per cent higher rque for one direction of rotation as against the other. elation of rotation to torque with amount of torque load differential can be specified by the designer to meet his specific requirements.

## Chrome-Moly Electrode

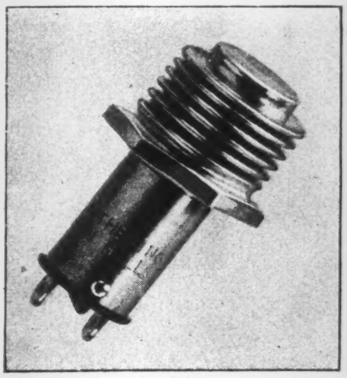
GIVING 8 to 10 per cent chrome, and 1.5 molybdenum weld deposit, the new alloy electrode, known as Chromend 9-M, developed by Arcos Corp., 1515 Locust street, Philadelphia 2, offers high strength and high corrosion resistance, combined with high creep resistance for high temperature corrosion resistance work.

## Ground Thread Socket Screws

NEW socket setscrews with centerless ground threads have been added to the line of Parker-Kalon Co., 200 Varick street, New York 14, and will be made available shortly. Threads of this setscrew are ground on hardened blanks, increasing strength, and are free from nicks, burrs and hardening cracks. A uniform Class 3 fit is assured. Ground thread screws, which heretofore have been made at high cost for precision equipment, will be equal in quality and sell at a price no higher than ordinary cut thread socket setscrews. Equipment is now being installed to produce the new screw in all standard sizes and when sufficient stocks are built up, will be released for sale.



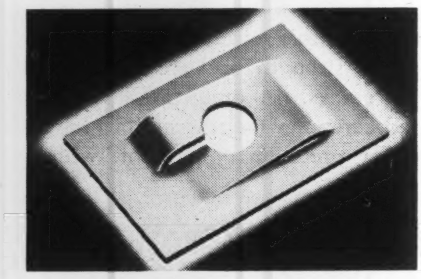
## "Environmentproof" Switches



AN ENVIRONMENTPROOF switch for use in damp or wet places, or where dirt, dust or fumes are present, has been announced by Robert Hetherington & Son Inc., Sharon Hill, Pa. Two methods to make these switches "environmentproof" have been worked out by the company, one utilizes a bellows on the plunger end of the switch, and the other a rubber boot over the plunger. In temperatures down to –40 degrees Fahr., the rubber boot has proved satisfactory under many conditions. However, where hydraulic fluids and gasoline or other solvents are present, or where switches encounter extremes of heat and cold, metal bellows are recommended. The switches, for both industrial and aircraft use, can be furnished with total movement of 7/16-inch. They are completely sealed, back and front, and are made for alternating or direct current. All contacts are double-break type, solid silver.

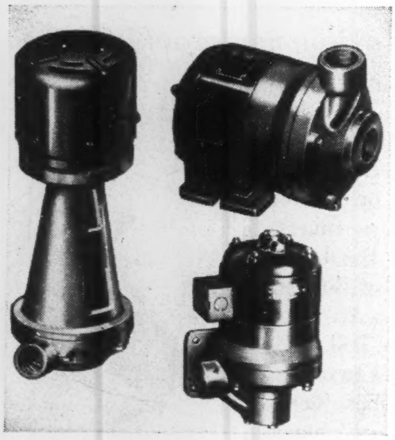
## *Self-Locking Sheet Metal Fastener*

MEETING ALL requirements of the AAF Specifications No. 25533, the new-type, self-locking sheet-metal fastener introduced by Adel Precision Products Corp., Burbank, Calif., is a one-piece, resilient, spring steel stamping. It grips screw threads all around with full 360 degree engagement. No lock washers or bearing washers are required. The new nut is available in a variety of types and sizes for sheet-metal screws and also may be formed as an integral part of assemblies.



## *High-Volume, Centrifugal Pumps*

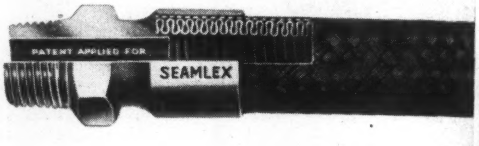
FEATURES OF the new line of Superflo centrifugal pumps, recently introduced by Gray-Mills Co., Evanston, Ill., include high volume of delivery per horsepower and ease of maintenance. Heavy-duty motors are mounted integrally to the pumps for compactness and strength, and are available in sizes of 1/25, 1/8, 1/4 and 1/2-horsepower. Capacities of the pumps range from 9 to 75 gallons per hour. There are 13 basic models, some of which are for horizontal-external mounting, for vertical-external mounting, and for submerging in the coolant system reservoir.



This type of pump has floating impellers, and may be us for fluids having abrasive content. Grease-sealed mo bearings and mechanical seals of pumps require no lub cation.

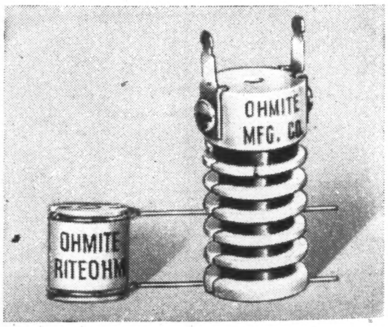
## *Tubing for High Temperatures*

EQUIPPED WITH "W" fittings, the new tubing offer by Seamlex Co. Inc., Long Island City, N.Y., is suitab where vibration and medium internal pressures at a temperature commensurate with the heat-resisting qua ties of the metals are involved. All parts—fittings, brai sleeves, and tubing—are welded or brazed into a sing unit. Fittings are of sturdy design, for continuous, al around service. Braid-protective sleeves are heavy-gag seamless tubing. Bell-shaped contours on the inside of the free ends prevent damage to braids.



## *Precision Resistors Offered*

TWO NEW series of Riteohm precision resistors, Seri 82 and 83, have been announced by Ohmite Mfg. Co 4835 Fluornoy street, Chicago 44. Mounted with through-bolt, the Series 82 resistor has two lug termina at one end fastened by screws, while the Series 83 ha radial wire leads. Specially enameled alloy resistance wi is noninductively pie-wound on a nonhygroscopic ceram bobbin which has a hole through the center for a No. screw. After being wound, units are vacuum impregnate with a special varnish, providing insulation and protectio against humidity. To make the units particularly suite for the tropics, they can be supplied with a varnish coa ing. Series 82 is available in three sizes: 11/16 x 1 1/8 1 7/16 or 1 3/4 inches for the 2, 4 and 6 pie units, re spectively. Minimum resistance is .1-ohm for all units an the maximum is 400,000 ohms for the 2-pie unit, 750,00 ohms for the 4-pie, and 1 megohm for the 6-pie unit. Als available in three sizes, Series 83 is 1/2 x 7/16 x 5/8 or inch in size. The first two units in this series are 2-pie whi the third is a 4-pie unit. Minimum resistance is 10 ohm



# Low Cost

What is the real cost of a bearing? Is it the purchase price alone? The length of service and the kind of performance delivered? Does the installation—maintenance and repair costs enter into it?

The real cost includes all of these . . . and more.

Postwar competition will force all manufacturers to deliver the highest quality at the lowest price. Your first step, when specifying bearings, is to be absolutely sure of the type. There is *one* correct bearing for each application. The bearings should be manufactured and finished in such a way as to cut installation costs to a minimum They should be interchangeable Finally, the quality should enable you to guarantee the length of service . . . in years . . . with a minimum of attention.

Our many years of wide experience enables us to give good sound advice to manufacturers of all types of products. Our manufacturing facilities plus our skilled personnel enables us to produce bearings strictly according to specifications. *Now* is the time to consult with us on your future bearing requirements. Remember—the most expensive bearing in the world . . . is the one that fails.

**JOHNSON BRONZE CO.**

**525 S. MILL STREET • NEW CASTLE, PA.**

VICTORY
COMES FIRST
*All our present production is for Armament*

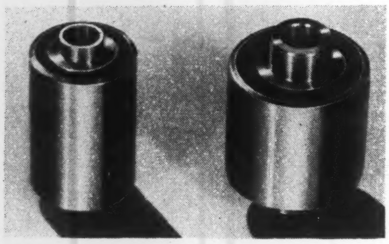
**JOHNSON**
**SLEEVE BEARING HEADQUARTERS**
**BRONZE**

BRANCHES IN
18 INDUSTRIAL
CENTERS

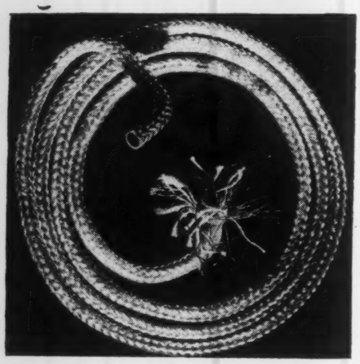
for all units and the maximum is 200,000 ohms for the small 2-pie unit, 400,000 ohms for the large 2-pie and 800,000 ohms for the 4-pie unit. A few common applications for the resistors include voltmeter multipliers, laboratory equipment, radio and electrical test sets, attenuation pads, and electronic equipment.

## *Insulated Cowl Mountings*

FIRST OF A series available and approved for military aircraft, new insulated cowl mountings introduced by Bushings Inc., 3442 West Eleven Mile road, Berkley, Mich., are impervious to decomposition by oil. They have a high rate of vibration and shock absorption, and a high resistance to heat. Assembly of metal inner and outer shells with synthetic is accomplished without vulcanizing. Bond is mechanical rather than chemical, permitting the use of any combination of rubber or synthetics with materials such as plastics, wood, steel, brass, powder bearing materials, etc.



## *Radio Shielding Sealing Gasket*



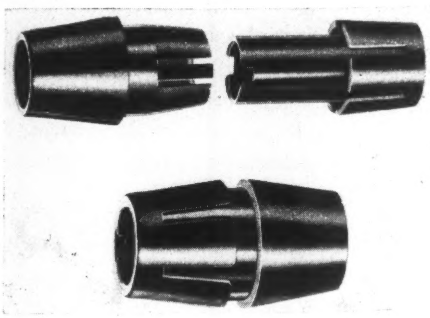
INTRODUCED by B. F. Gladding & Co. Inc., South Otselic, N.Y., a new radio shielding sealing gasket eliminates radio and radar interference caused by high tension currents escaping through parting joints in aircraft ignition systems, in all radio-frequency ranges from 200 to 300,000 kilocycles. Outer circumference of the gasket will conduct an electrical current and provide an adequate moistureproof seal. It is also effective in maintaining a pressure seal on supercharged ignition systems. According to the company the gasket has many applications.

## *Tie Rods Absent in Air Cylinders*

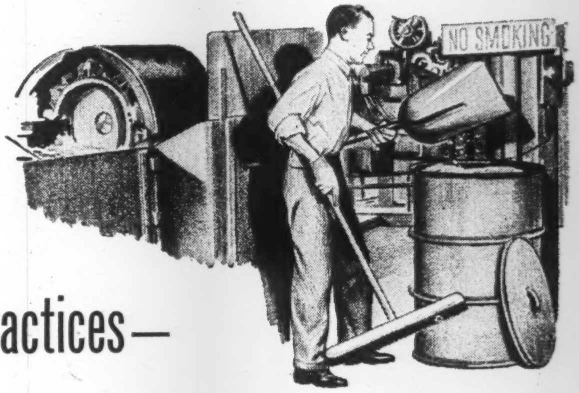
ABSENCE OF tie rods is an outstanding feature of the new double-acting nonrotating air cylinders announced by Gerotor May Corp., Logansport, Ind. A keeper ring instead assures leakproof construction and permits co to be rotated to any position for convenient locatio pipe connections. This construction also permits more c pact design and allows the cylinder to be applied i minimum space. Suitable for 150 pounds per square i air pressure and 300 when applied to oil and water s vice, the cylinders are available in seven standard mo ings, in twelve diameter bores and in any length of str Those used for oil-hydraulic service are available w auto-type piston ring construction. Self-adjusting comp tion packings seal the cups and piston rod. Other featu include large pipe connections, ample bronze rod gui oversize piston rod, heavy-duty piston assembly and hea duty alloy covers.

## Blind Plastic Grommets

FOR ELECTRICAL insulation applications of all typ nonflammable plastic blind grommets produced by Vict Mfg. Co., South Pasadena, Calif., are composed of t parts molded from a special Lumarith. These two pa are so designed that they may be applied from one s only by means of a special tool. To install the gromm it is slipped on a special tool and thrust through a hole the partition. As the grommet is drawn together, an u dercut section on one half is engaged with a spring lo ing section on the other, the tool then forcing the two s tions together and locking them into one integral u The grommets are furnished in a wide range of sizes to commodate cables and tubes from ¼-inch through 2 inc in diameter. Two sizes for wall thickness are suppli one to accommodate ⅛-inch to ½-inch and the second bulkheads ½-inch to 1 inch in thickness. Colors, availa in suitable plastic materials, are utilized for decorative fects and for identification purposes. Clear grommets m



# Fabricating magnesium —like other metals— requires only simple precautions



## Good housekeeping— plus established magnesium shop practices— keeps production moving

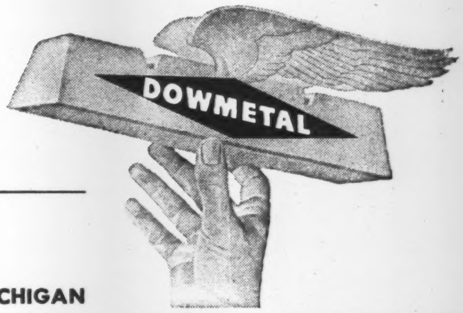
tting best results in magnesium fabrication has en a highly productive Dow research program for ood many years. Not only in Dow's own complete ops, but also in the many other plants that use gnesium as a standard production metal, operations e speeded by following simple, well-defined shop ocedures.

is largely involves "good housekeeping"—the type plant practice that modern manufacturers observe a matter of course: machines and floors should be pt reasonably clean; sensible clothing should be worn; approved fire extinguishers should be kept on hand.

Beyond that, specific techniques—quite similar to those used with other metals—expedite the manufacture of magnesium products in the full variety of common fabrication methods.

These procedures . . . as they apply to magnesium fabrication in your own shop . . . are available to you from Dow technicians thoroughly versed in the field. Call the nearest Dow office for consultation.

**MAGNESIUM**



*THE METAL OF MOTION*

**GNESIUM DIVISION, THE DOW CHEMICAL COMPANY, MIDLAND, MICHIGAN**

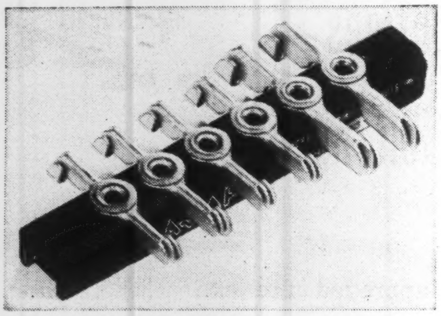
York • Boston • Philadelphia • Washington • Cleveland • Detroit • St. Louis • Chicago • Houston • San Francisco • Los Angeles • Seattle be used to facilitate inspection. Fields of application include insulation in electrical, plumbing, air conditioning, automotive, shipbuilding and aircraft industries.

## *Thermoplastic, Transparent Tape*

**T**RANSPARENT, flexible, thermoplastic tape, introduced by Irvington Varnish & Insulator Co., 6 Argyle Terrace, Irvington, N. J., facilitates inspection and servicing of equipment on which it is used. Useful not only as electrical insulation, Fibron Tape No. 3 also protects wiring, cables and equipment against abrasion. It is heat-sealing, flame-resistant, flexible at low temperatures, and resists attack by acids, alkalies, moisture, oil, grease and corrosive fumes. With proper adhesives, it may be bonded to fabrics, metal, ceramics, wood and other materials. According to the company, the tapes have been applied successfully as gasketing material, in construction of automotive and aircraft lighting and ignition harnesses, and in splicing of plastic-insulated wire and cable.

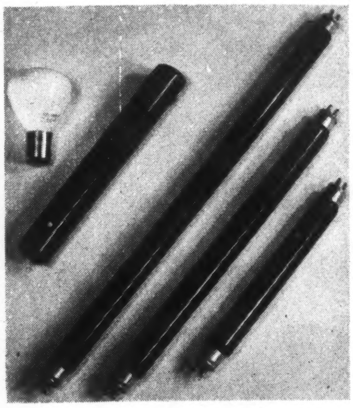
## *Solderless Connector Strips*

**E**SPECIALLY ADAPTED to hangers for sectional wiring, a simplified connector strip manufactured by Aircraft-Marine Products Inc., 15910 North Fourth street, Harrisburg, Pa., is for use with the company's solderless knife-disconnect splicing terminals. It affords quick assembly of any number of terminal connections on a single block of insulating plastic. Integral members are firmly riveted to the plastic base. Any knife-switch connect end may be specified to accommodate any wire size from 22 to 10 inclusive and, where required, the opposite end may be made for a different wire size. Permanent or temporary shorting connections can be incorporated between two or more adjacent terminals.



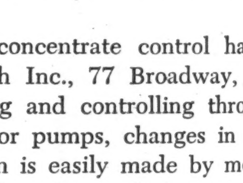
## *Black Light Lamps*

**F**OR INDUSTRIAL, aircraft and marine instrument illumination, as well as many other applications, black-light lamps are now available from Sylvania Electric Products Inc., 500 Fifth avenue, New York. They are furnished for operation on 120-volt alternating-current and 24-28 volt direct-current circuits. The near ultraviolet RP12, rated at 4 watts, is designed for use with a visible light filter where a lightweight, compact lamp is required. Equipped with a polarized bayonet base, it may operated in any position for use with aircraft ins ments, inspection or other industrial equipment. T red-purple series of tubular lamps require no visible li filter and operate with standard fluorescent-lamp acc sories on 120-volt alternating-current circuits. They supplied with miniature bi-pin bases in five sizes, rang from the 6-inch T5 rated at 4 watts, to the 36-inch rated at 30 watts.

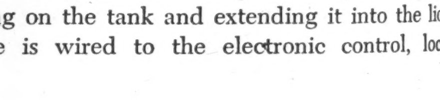


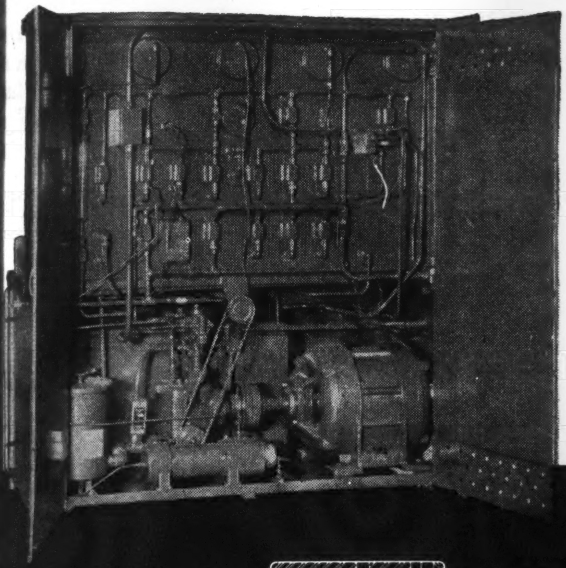
## *Electronic Liquid Control*

**A**N ELECTRONIC concentrate control has been veloped by Photoswitch Inc., 77 Broadway, Cambri 42, Mass., for detecting and controlling through ope tion of signals, valves or pumps, changes in liquid c centrations. Installation is easily made by mounting t probe fitting on the tank and extending it into the liqu This probe is wired to the electronic control, locat wherever desired. An adjustment on the control housi is set so that the control relay will operate when liq of a predetermined electrical resistance contacts the pro While the unit is immersed in liquid of any other sistance, the control remains inoperative, but when change in concentration alters the conductivity of liquid to the necessary degree, the electronic control



PHOTOSWITCH
ELECTRONIC
CONCENTRATE
CONTROL
TYPE P25N

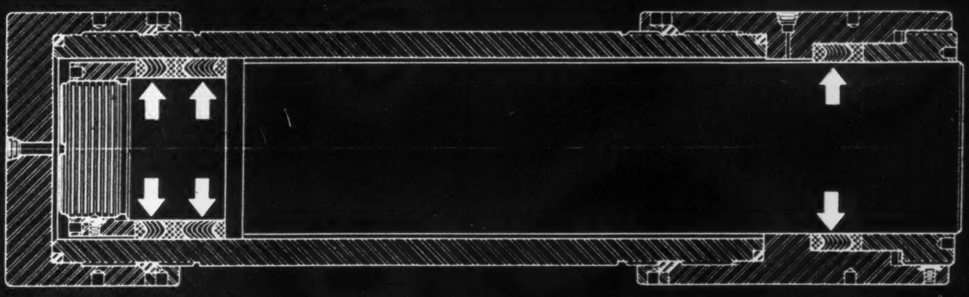




# 20,000 PSI in this AAF test machine . . .

Photo and drawing courtesy Greer Hydraulics, Inc.

Rear view of Greer test machine showing booster cylinder at bottom. Design at right illustrates essential details of this cylinder.



## AND VIM PACKINGS MADE IT POSSIBLE!

nfinitely variable pressure control, from ero to 20,000 lbs., is provided in the above st machine built for the U. S. Army Air orces. The main pumping circuit enables 000 PSI by dynamic loading, then static oading in the booster cylinder can be pped to the 20,000 mark.

IM Leather "V" Packings of the type origi-ally developed by Houghton and resin-npregnated for aircraft hydraulic mech-nism, hold the seal at all times. In fact, he manufacturer tells us that he is "very rateful to E. F. Houghton & Co. for the evelopment of this type of packing, as without it we could not have a 20,000-lb. pressure system."

VIM "V" Packings have held up under all tests in the machine pictured (rear view) above. And they'll hold up for you . . . and you . . . and you, who design and make hydraulic equipment. With them goes a complete engineering service that helps plan designs that will work. May we work with you on packing problems?

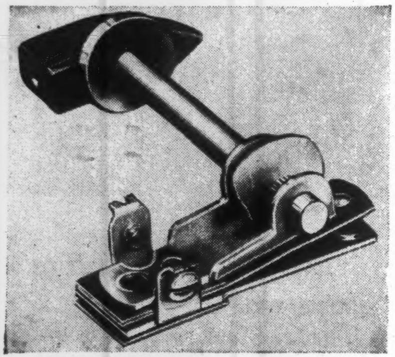
**E. F. HOUGHTON & CO.**
PHILADELPHIA
*Chicago* *San Francisco* *Detroit*

**HOUGHTON'S Engineered VIM Leather Packings**

lay is energized to operate signals, valves or pumps. Type P25N control is recommended for detecting of contamination in boiler condensate, or for use wherever it is necessary to maintain an interface between two liquids differing slightly in electrical conductivity. This type has a sensitivity range of 100 to 5000 ohms and operates on a 5 per cent change in probe-circuit resistance. The voltage applied to the probe never exceeds 25 volts, eliminating possibility of electric shock or explosion hazard. Type P25N incorporates a single-probe, double-throw relay, rated at 10 amperes alternating current, 5 amperes direct current, for normally closed and normally open operation, and requires a supply of 115 volts alternating current, 60 cycles.

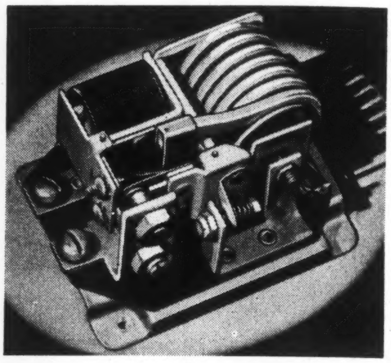
## Cam-Adjusted Thermostatic Switch

**A**NNOUNCED BY George Ulanet Co., 88 East Kinney street, Newark, N. J., the new dual cam-adjusted thermostatic switch can be used for either or both a thermostat and a manual switch. It eliminates the necessity of an additional on and off switch and permits instant adjustment of the thermostat. Simple and foolproof in operation, the thermostatic switch is suited for mounting on the bottom of electrically heated equipment such as glue pots, hot plates, etc., where manual on and off knob is desired. Knob can be located any distance from the thermostatic element by lengthening cam shaft. Pointer on the knob can be set at any desired temperature for control of the heating element. The unit is available in three adjustable temperature brackets: 50 to 300 degrees, 50 to 450 degrees, and 50 to 700 degrees Fahr. It is rated at 1500 watts, 115-230 volts alternating current.



## Reverse Current Relay

**M**ANUFACTURED BY R-B-M Mfg. Co., Division of Essex Wire Corp., Logansport, Ind., the new Type 9100 reverse current relay, originally for special use on auxiliary engine-driven power generators in Army tanks and airplane bombers, is said to be equally effective for any "tough" low-voltage direct-current application. Sturdy and compact in design, the relay offers dependable operation on either stationary or mobile equipment. A magnetic latch prevents accidental closing of armature and contacts due to vibration up to 10g, or heavy shock. Contacts are rated at 100 amperes at 30 volts direct current maximum. Physical dimensions of the relay are: Wi 4 1/16 inches; depth, 3 7/16 inches; and height, 2 1/ inches. Weight is approximately 1.6 pounds. This verse current relay is also available without magnetic la (Type 9000) for use where severe vibration and shock not encountered, in sizes as low as 300 watts at 6, 12, and 24 volts direct current.

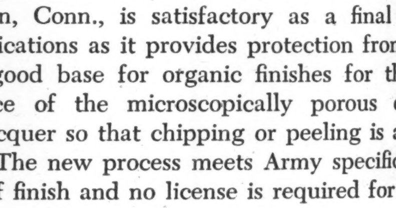


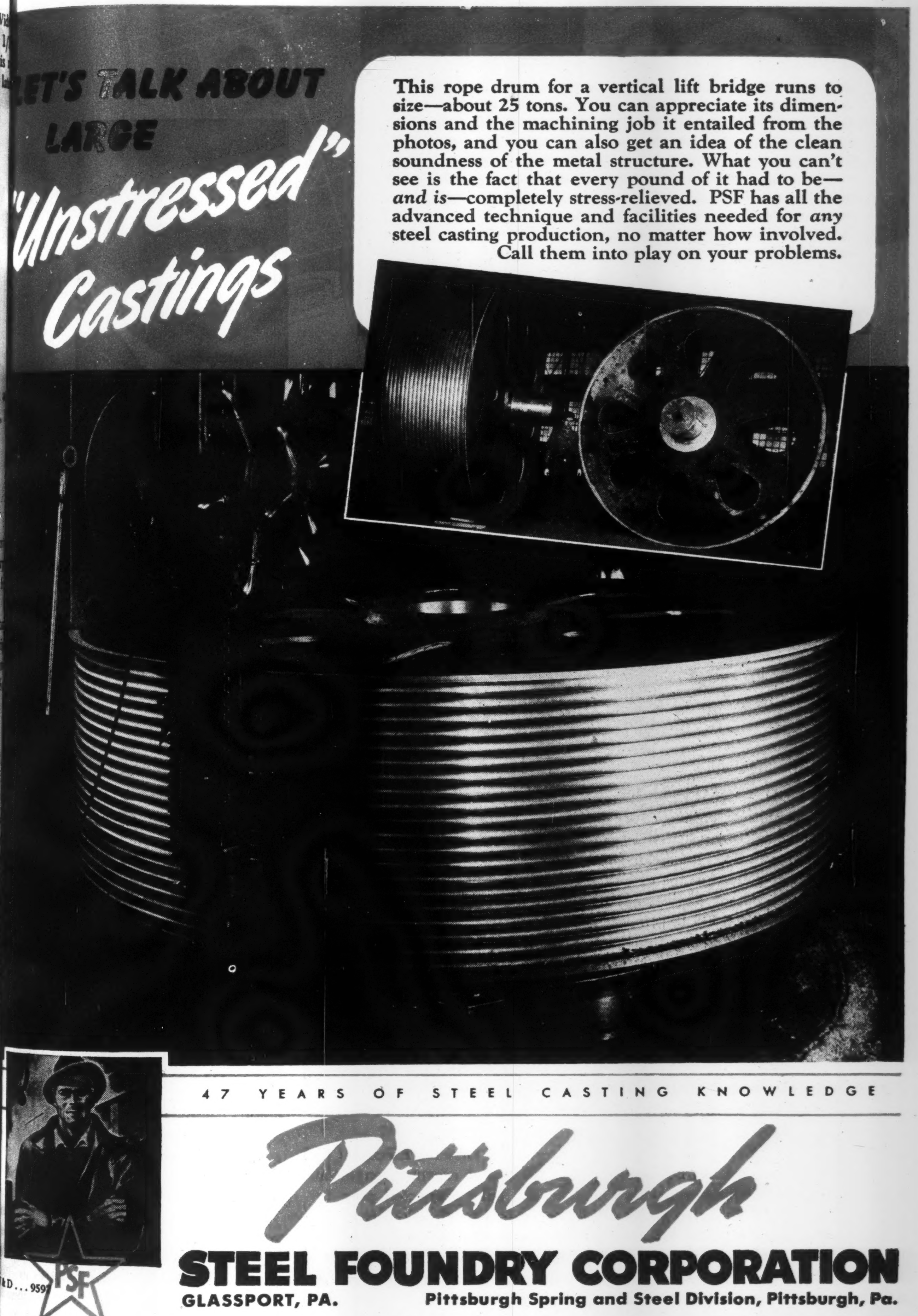
## Phosphate-Type Finish

**O**F PHOSPHATE type, the new Phosteel finish for i and steel, announced by The Du-Lite Chemical Co Middletown, Conn., is satisfactory as a final finish many applications as it provides protection from rust. is also a good base for organic finishes for the cryst line surface of the microscopically porous coat gr paint or lacquer so that chipping or peeling is almost i possible. The new process meets Army specifications f this type of finish and no license is required for its use.

## Paper-Dielectric Capacitors

**H**ERMETICALLY-sealed, fixed paper-dielectric cap citors have been made available by General Electric C Schenectady, 5, N. Y., with glass terminal insulators, case styles CP-60, 62, and 64, characteristics E and Meeting the Proposed Joint Army-Navy Specificati JAN-C-25, they are for use in combat communicati equipment where severe operating conditions may encountered. Designed to provide a high degree of r sistance to humidity, fungus growths and termites, t capacitors have a unique construction which does n rely on matched coefficients of expansion of glass an metal to resist thermal shock, nor on soft solder for m chanical strength. All three case styles are availab in both single-section and two-section construction. C pacitance values range from .05 to .5-microfarad, for vol ages of 600, 1000 or 1500 volts.





This rope drum for a vertical lift bridge runs to size—about 25 tons. You can appreciate its dimensions and the machining job it entailed from the photos, and you can also get an idea of the clean soundness of the metal structure. What you can't see is the fact that every pound of it had to be—*and is*—completely stress-relieved. PSF has all the advanced technique and facilities needed for *any* steel casting production, no matter how involved. Call them into play on your problems.

47 YEARS OF STEEL CASTING KNOWLEDGE

**Pittsburgh**

**STEEL FOUNDRY CORPORATION**

**GLASSPORT, PA.** **Pittsburgh Spring and Steel Division, Pittsburgh, Pa.**

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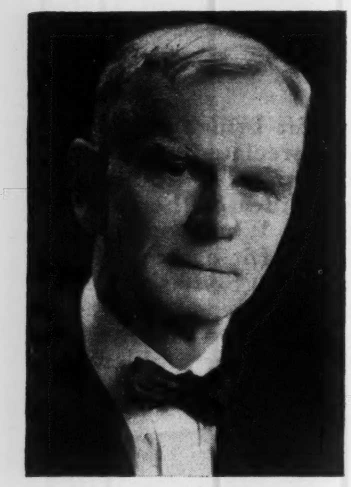


Laurence A. Hawkins

Dudley E. Chambers

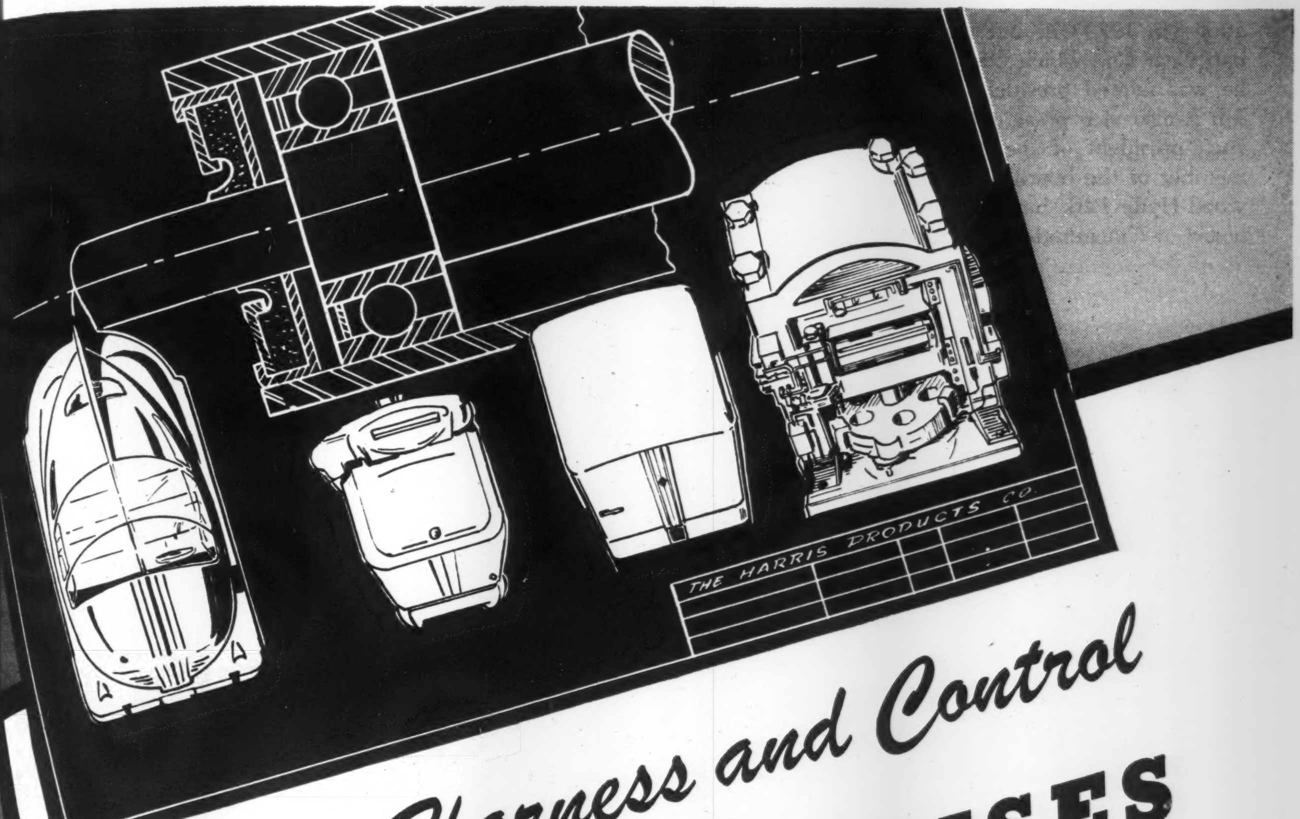
Paul W. Christensen

# MEN... of machines

**L**AURENCE A. HAWKINS has retired as executive engineer of the General Electric Co. research laboratory which position he held since 1912. His associations in the electrical industry began in 1899 in the engineering and patent department of the Stanley Electric & Mfg. Co., Pittsfield, Mass. In 1903 he joined the General Electric Co. as patent attorney, later transferring to signal work in the transportation engineering division. He was transferred in 1912 to the research laboratory, first as engineering assistant to the director, then as executive engineer. Dr. Hawkins was born at Pittsfield, Mass., in 1877, and was graduated from Williams college in 1897 and from Massachusetts Institute of Technology in 1899. Williams college conferred its honorary degree of Doctor of Science on him in 1944. As executive engineer of the research laboratory, his work assumed particular importance when practically all of the laboratory became engaged in war research.

**D**UDLEY E. CHAMBERS, who has been with General Electric Co. since 1928, has replaced Dr. Laurence A. Hawkins as executive engineer of the research laboratory, with which he had been connected prior to his appointment. Though a native of Missouri, Mr. Chambers received his education in the schools of California, graduating in 1927 with the degree of A.B. in electrical engineering from Leland Stanford university. The following year, after receiving the degree of electrical engineer, he joined the General Electric Co. at Schenectady. After being employed in the research laboratory with a group working on electron tubes, he entered the company's test course. Up completing this course, he was tra ferred to the industrial control divisi of which he became head in 1933. F years later he was appointed assist engineer of the transmitter engineeri division, where he remained until 19 when he returned to research.

**P**AUL W. CHRISTENSEN, preside and general manager of the Cincinn Gear Co., has recently been elect president of the American Gear Man facturers association. For a number years he has been active in the associ tion's activities, and recently has co pleted serving a year as vice preside As a member of the association's boa for a number of years, he has taken major part in the guidance of its affai His entire business career has been wi the Cincinnati Gear Co., which was ganized by his father. He was grad ated from Purdue University in 19 and received his mechanical engineeri degree three years later. Upon grad


THE HARRIS PRODUCTS CO.

# How to Harness and Control VIBRATION NOISES

Today, vibration control on postwar equipment is a major subject with most of the leading equipment and appliance manufacturers because it adds greatly to the longevity, efficiency and performance of their products. Vibration control also means quiet, smooth operation which in turn is a potent sales point in selling appliances of all kinds.

Harris *Torflex* Flexible Bearings will prolong life, lower maintenance costs, insure more dependable and accurate operation and increase efficiency. This is done by absorbing shock, controlling vibration, silencing noises, providing flexibility for cushion and misalignment, transmitting torque and eliminating wear and friction. *Torflex* Bearings take axial, torsional, and radial loads and come in a wide range of sizes, carrying loads from ounces to tons. They are widely used and universally approved.

Harris pioneered in the field of engineered vibration control, consequently *Torflex* Flexible Bearings and Harris Mounts are widely used on prewar equipment, Army and Navy planes, military mobile units of all kinds, industrial equipment, automobiles, electrical and electronic products. If you have a vibration problem, our engineers would welcome an opportunity of working with your engineers, just drop us a line.

**HARRIS** PRODUCTS COMPANY
CLEVELAND 4, OHIO
U. S. A.

ation (in 1923) he became vice president of the Cincinnati Gear Co., which position he fulfilled until 1941 when he was named president of the company. Mr. Christensen is also vice president of the United Aircraft Products Inc., president of the American Precision Products Inc., member of the board of Doyle Mfg. Co., director of Norwood-Hyde Park bank & Trust Co., and chairman of the board of Cincinnati Steel Treating Co.

**HAROLD M. DETRICK**, well known in the radio apparatus design field, brings twenty years of experience to Bendix Radio in his recent appointment as chief engineer of the broadcast receiver section. In 1925 he started in radio engineering with the Metro Electric Co., where he gained experience in vacuum tube design and manufacture. Five years later he joined the engineering staff of the U. S. Radio & Television Co., designing many models of home radio receivers. In 1927 he accepted a position of assistant chief engineer at the Silver Marshall Co. where he contributed many advancements in radio receiver design. While there he also designed public address amplifier and loud speaker equipment, amplifiers and microphone mixing panels for broadcast station studios. In 1931 he helped organize McMurdo Silver Co. Inc., and as its chief engineer helped design a custom-built line of broadcast receivers. Mr. Detrick also was instrumental in establishing the Hallicrafters Corp. where he assumed the post of vice president and chief engineer. He joined the Clough-Brengle Co. in 1934 as chief engineer. Under his direction oscilloscopes, signal generators and other testing instruments, amplifiers, and photoelectric exciter lampheads for sound-on-film motion pictures were designed. In 1936 he was appointed assistant chief engineer of the Stewart-Warner home radio receiver division where for more than five years he was engaged in the design of all kinds of radio receivers. From 1942 until his present appointment he was chief engineer of General Instrument Co. in charge of engineering design of apparatus for the Services.

◆

**L. A. Majneri has been advanced from chief engineer to vice president in charge of engineering of Warner Aircraft Corp., Detroit, while W. A. Wiseman, formerly assistant chief engineer, becomes chief engineer.**

◆

**R. W. Waring recently resigned as materials engineer of Sperry Gyroscope Co. Inc., to become chief engineer of the East Main plant of Bridgeport Brass Co.**

◆

**George Steven has been appointed executive engineer of the Buffalo Works of Worthington Pump & Machinery** Corp. **Harold W. Whiting**, formerly Mr. Steven's sistant, now succeeds him as chief engineer of Buff Works Compressor division.

◆

**William F. Clark** has joined Weinman Pump & S ply Co., Pittsburgh, as chief engineer. He formerly been lubricating and hydraulic engineer, United Engine ing & Foundry Co., Pittsburgh.

◆

**Harry E. Dunham** has been appointed manager of patent department, General Electric Co., Schenecta N. Y. **Harry R. Mayers** succeeds Mr. Dunham as sistant manager of this department.

◆

**Edward J. Charlton**, since 1942 assistant to the p ident of Lukenweld Inc., has been appointed manager, velopment engineering, for Lukens Steel Co. and its sidiaries.

◆

**Edward E. Kuphal**, since 1942 assistant director of search in charge of chemical engineering and mechani design, has been named director of manufacture for Ge eral Mills Inc.

◆

**Alfred Iddles**, special assistant to the vice preside in charge of engineering, recently was elected as one the vice presidents of The Babcock & Wilcox Co. H duties will be extended to include supervision of certain the company's engineering activities.

◆

**John A. Comstock** has recently been appointed direct of research and metallurgy for all divisions of H. K. Port Co. Inc.

◆

**Allan M. Cameron** has been named general manag of equipment development and manufacturing, Cont nental Can Co. In addition to his development work h will have jurisdiction over the company's machine shop and will maintain his headquarters at the McDonald M chine Shop in Chicago. Mr. Cameron is also president the Cameron Can Machinery Co.

◆

**Dave Smith** has accepted a position in an engineerin capacity with the Western Gear Works. He will be con cerned with engineering of the company's pumping unit speed reducers, etc.

◆

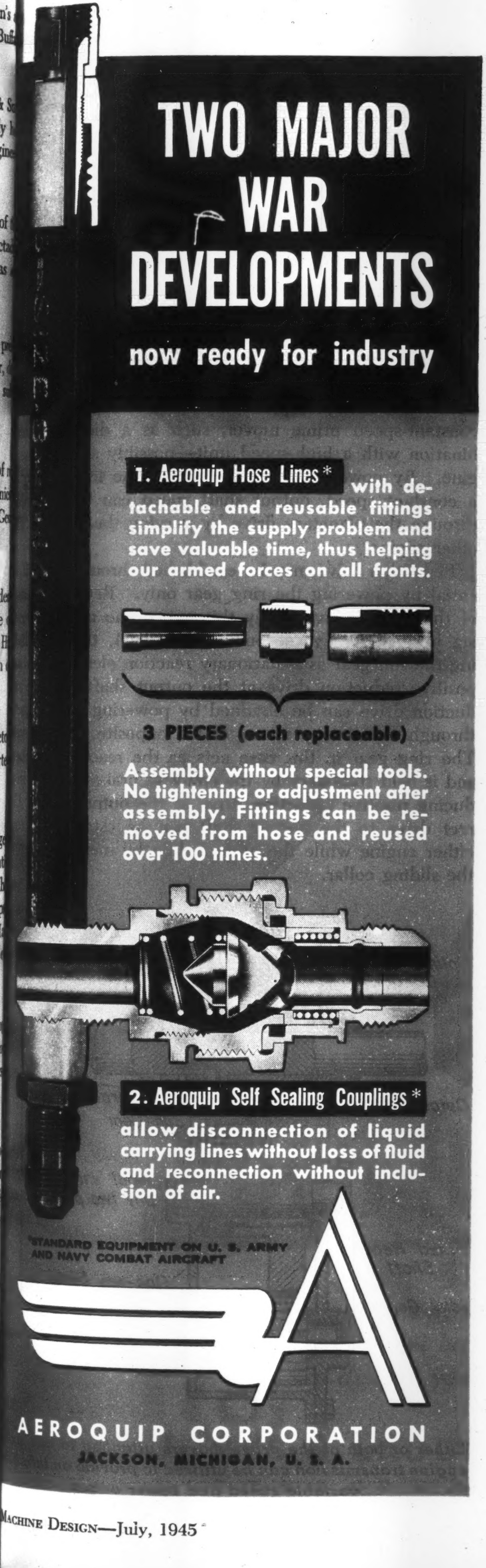
**William S. James** has been appointed director of auto motive research for the Ford Motor Co. This new divisio will function as a part of the company's automotive engi neering department under **R. H. McCarroll**, the com pany's executive engineer.

◆

**John P. Gobetti** is now manager of the mechanical de sign department of the Goodyear Aircraft Corp., Akron, O He left his position as design engineer of Glenn L. Marti Co., to accept his new duties with the Goodyear company

◆

**V. L. Maleev** has been made consulting mechanica engineer with Harvey Machine Co. Inc., Los Angeles. H formerly had been a civilian employee of the U. S. Nava Engineering Experimental Station, Annapolis, Md.



TWO MAJOR WAR DEVELOPMENTS

now ready for industry

**1. Aeroquip Hose Lines*** with detachable and reusable fittings simplify the supply problem and save valuable time, thus helping our armed forces on all fronts.

**3 PIECES (each replaceable)**

Assembly without special tools. No tightening or adjustment after assembly. Fittings can be removed from hose and reused over 100 times.

**2. Aeroquip Self Sealing Couplings*** allow disconnection of liquid carrying lines without loss of fluid and reconnection without inclusion of air.

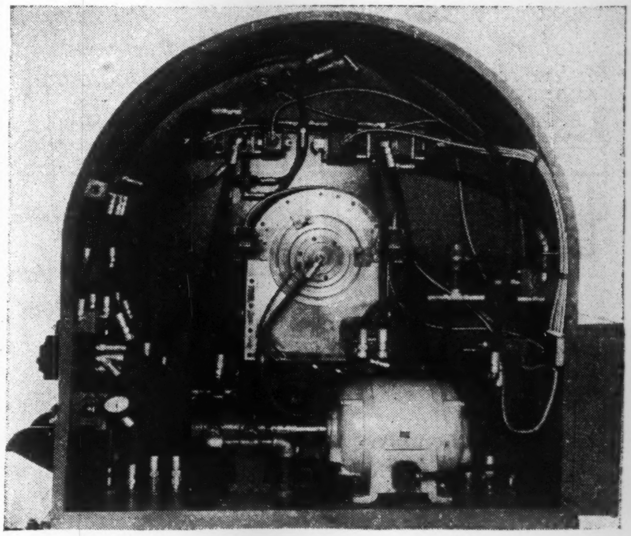
*STANDARD EQUIPMENT ON U. S. ARMY AND NAVY COMBAT AIRCRAFT

AEROQUIP CORPORATION
JACKSON, MICHIGAN, U. S. A.

# MACHINE TOOL PROBLEM SOLVED

The problem of positioning hydraulic lines in the design of machine tools, and later problems of maintenance accessibility and replacement, are solved by the use of a new type of flexible Hose Line with Detachable Fittings on machine tool installations.

The simplicity of using these flexible lines in place of rigid lines not only eliminates many questions of design layout previously necessary to accommodate fixed lines, but also permits complete assembly of the machine itself, followed by the subsequent application of these flexible lines, accommodating themselves to any space available, over or around various parts.



The above view illustrates the rear of a special milling machine with three hydraulically driven milling heads. Aeroquip hose lines are handily adapted to all positions and connections where rigid tubing would have required more extensively engineered outlet and space arrangements. At the same time performance under extreme vibration has been greatly improved and failures or ruptures are practically eliminated.

Aeroquip hose lines, which are "AN" standard on U. S. Army and Navy aircraft, are suitable for all hydraulic actuation, including installations for lubrication, pressure switches and power transmission.

The detachable fitting feature simplifies maintenance, permitting quick attachment, and continued re-use of the individually interchangeable fitting parts with new hose lines "tailored to the job" on the spot by simply cutting new Hose to the desired length.

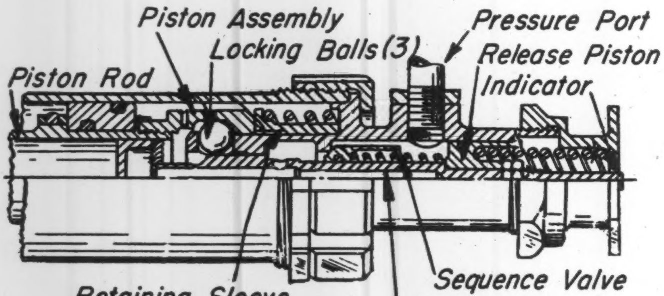
The adoption of flexible Hose Lines to Machine Tool installations is gaining rapid recognition among machine tool designers and builders.

# Noteworthy Patents

## Hydraulic Cylinder Has Safety Lock

For hydraulically operated mechanisms, a cylinder that automatically locks in normal off-power position is covered by patent 2,360,535 recently assigned to the Lockheed Aircraft Corp. This cylinder retains any operated mechanism in fixed position obviating accidental movement in case of undue shock, leakage or failure of the hydraulic power unit.

As shown in the accompanying drawing of the cylinder, three balls floating in fixed slots engage an extension of the piston assembly. A sliding plunger, spring loaded, holds the balls in this locking position to assure positive retention of the loaded piston rod.



***Piston locking mechanism in this hydraulic cylinder positively holds loaded piston in position***

Hydraulic pressure entering at the indicated pressure port first releases the piston and ball assemblies and then effects movement of the piston to the left. To assure positive break of the lock prior to release of hydraulic pressure into the piston, a sequence valve is used. This valve prevents pressure from passing into the cylinder until the locking plunger first releases the balls. Full travel of the locking plunger-release piston opens the sequence valve to effect free flow into the cylinder. When pressure is transferred to the opposite port, a spring-loaded retaining sleeve automatically holds the balls in release or inoperative position until the main piston assembly again reaches extreme right position. The locking mechanism can be designed for operation at either extremity of piston travel.

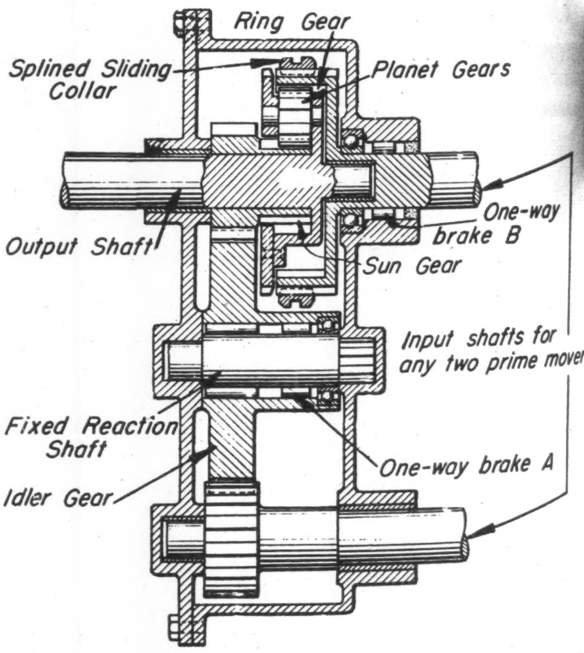
## Transmission Combines Power

Any two prime movers or combination of prime movers—diesel, gasoline, electric, or steam—can be utilized to obtain their combined power output at any desired speed ratio by means of a novel differential transmission shown in the accompanying illustration. Covered by patent 2,366,646 recently assigned to the Borg-Warner Corp., this transmission effects such a combination of power regardless of the relative values of the power d livered by the prime movers used.

Power from each of the engines or motors is delivere to the transmission through a hydraulic coupling and standard friction-type clutch. Direct drive with a rig speed ratio between the input shafts and the outp shaft is accomplished by a sliding collar which locks o the differential gearing. Any variations in engine spee are absorbed in the fluid couplings.

Controlled output speed is easily obtained by using on constant-speed prime mover, such as a diesel, in com bination with a high-speed unit—possibly a gasoline en gine. By powering the ring gear of the transmission a a constant speed, output shaft speed can be controlle through the planetary drive by varying the speed of th other engine.

Reduction drive may be established through the differ ential by powering the ring gear only. Reverse rotatio of the sun gear with only this one prime mover opera ing is obviated by the one-way brake *A*, thus establish ing the sun gear as a stationary reaction element creatin positive reduction drive at the output shaft. A like re duction drive can be obtained by powering the sun gea through an idler by means of the opposite prime move The ring gear in this case acts as the reaction membe and is held stationary by the one-way brake *B*, again pro ducing positive reduction drive at the output shaft. Di rect drive to the output shaft can be established fro either engine while the other is idle by merely engagin the sliding collar.



***Either or both of the prime movers attached to this multi engine transmission can be utilized to provide an infinitely variable speed and power output***

# J&L STEEL

# COLD DRAWN BARS

*Precision, smooth finish, uniformity are the prime qualities you insist on in cold drawn steel bars. Our metallurgical engineers will be glad to discuss your production problems with you.*

**JONES & LAUGHLIN STEEL CORPORATION**
**PITTSBURGH 30, PENNSYLVANIA**

# ASSETS to a BOOKCASE

### Science of Clocks and Watches

*By A. L. Rawlings, temporary principal experimental officer, Admiralty Compass Department; published by Pitman Publishing Corp., New York; 246 pages, 6 by 9 inches, clothbound; available through* Machine Design, $3.50 *postpaid.*

Proper measurement of time still remains one of the most important things in the world today. Measures of speed, acceleration, force, heat, elasticity, electric current, voltage, etc., all contain a consideration of time. Into every machine the designer creates is built the element of time, whether he is totally aware of it or not.

Older than history itself, the subject of horology has long been looked upon as a craft requiring the exquisite manual skill of a gifted artisan. Scientific literature of timekeeping is scant and widespread, misconception often overlaying the few writings to be found on the subject.

Here is a book that clears away these difficulties and brings together thoughts and theories on the science of timekeeping. The author applies modern methods of scientific analysis to shed new light on the many old problems, buried in tradition for centuries, as he views long-held ideas with fresh, unhampered realism. Concerned primarily with the scientific principles of horology, the book deals with units of time, oscillatory motions, pendulums, escapements and impulse devices, escapement errors, balance-wheels and their movements, gear trains, friction and lubrication, etc. The many interesting and important developments of recent years in clocks for observatories, marine chronometers and gyro-compasses, are particularly timely. The proposed design for a new type chronometer and the details of the little-known 400-day clock will prove specially engaging to the reader.

With clear and readable style, the work is that of an engineer, inventor and experimentalist. It brings to the designer of machines—especially those involving mechanical timing—the fundamentals required for a thorough understanding of the mathematics of clock and watch timing mechanisms.

□ □ □

### Production Line Technique

*By Richard Muther, department of business and engineering administration, Massachusetts Institute of Technology; published by McGraw-Hill Book Co. Inc., New York; 320 pages, 5¾ by 8¾ inches, clothbound; available through* Machine Design, $3.50 *postpaid.*

Success of American competition in postwar markets will depend to a large extent on the development of more efficient production methods. Efficient manufacturing involves primarily the coordination of material, m and machinery in the easiest and most effective way. this end production-line methods of manufacture o the greatest future promise.

In this book the author presents accumulated d which represents firsthand practical experience and thoritative knowledge gathered from more than one h dred industrial organizations. His observations have b made as operator, organizer, and analyst on line-prod tion. For the engineer interested in the nature, adv tages, or limitations of production-line methods, th field investigations of current practices will provide an cellent basis for a thorough understanding of all the fact involved. Those already concerned with production-l manufacture may find the author's fundamentals for s cessful operation and pitfalls to avoid in setting up a p duction-line lend new insight to the practical applicati of these procedures.

Of special interest at the present time as reconversi begins to assume importance should be the chapters production-line flexibility. Dispelling the fear that chang and/or improvements in products kill the value and sa ings in such manufacture, the author shows that li production offers opportunity for the introduction of considerably greater amount of flexibility than is genera assumed. Flexibility in design, output, layout, equipme and personnel are examined individually to ascertain t maximum degree of diversification that is possible a the modifying effects each of the factors present.

The presentation is at once objective and penetrati well organized, and amply illustrated with drawings a photographs.

---

### Practical Design for Arc Welding
### Volumn II

Taking up where the first volume left off, this bo again brings to the designer a multitude of practical "id stimulators" in a well-planned pictorial presentatio Stressing fabrication of tubular members and connectio the author outlines in unique sketch form the accomplis ments of designers in the field of welded structures.

Conveniently included in this book are the standa A. W. S. welding symbols and drawings of typical weld joints, sections, etc. Each drawing indicates plainly t proper type of weld and joint to specify for the applicatio

New and unusual in its treatment of this complex su ject, this second volume of a series by Robert E. Kinkea consulting engineer on welding, is published by The H bart Bros. Co., Troy, Ohio. Containing 100 design plat 8½ by 11 inches, the book is clothbound and is availab from the Hobart company at $3.50.

## STAINLESS SURFACE HARDENING Case History #3

# Increasing Resistance to Erosion Caused by High Pressure Steam

IN the manufacture of turbines, Worthington Pump and Machinery Corporation specifies treatment of all valve stems, bushings, plugs and valve seats by the Stainless Surface Hardening Process. As a result, these vital parts of Stainless Steel are given an extremely hard case that satisfactorily resists the impact of high-velocity steam, superheated to 800°F. and backed by pressures up to 900 PSI. Worthington engineers also report an important saving in manpower . . . because surface-hardened parts require less time for finishing operation than other materials.

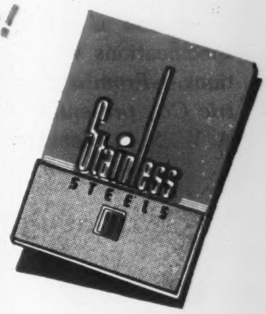
Efficient new process will extend life of your product's Stainless Steel parts!

*If you make or use Stainless parts, investigate the Stainless Surface Hardening Process. Gives up to 1100 Brinell surface hardness. Write or call today for full information. Forward some of your Stainless parts to us for sample treatment and quotations.*

## LET ONE CALL DO IT ALL . . . when you need **Stainless!**

**TROwbridge 7000**
**Teletype Cambridge 547**

Whatever your Stainless requirements, you'll receive same-day attention to your entire order from Industrial Steels, Inc. . . . for Industrial carries the largest and most diversified warehouse stock of Stainless Steels in America. Save time . . . save money . . . get action! Phone Industrial *first.* And if you need expert metallurgical counsel regarding any Stainless fabrication or specification problem, don't fail to request it. Send for catalog. INDUSTRIAL STEELS, INC., Phone TROwbridge 7000 or Teletype Cambridge 547, 250 Bent Street, Cambridge 41, Mass.



## STAINLESS SURFACE HARDENING CO.

**255 BENT ST., CAMBRIDGE 41, MASS.**

Subsidiary of Industrial Steels, Inc.

**Stainless Surface Hardening Co.**
**263 Bent St., Cambridge 41, Mass.**
**Gentlemen:**
**Please send complete details regarding the Stainless Surface Hardening Process.**

**Name**..............................................................
**Firm Name**.......................................**Title**..................
**Street**............................................................
**City**..............................................................
................................................**State**..................
**We are interested in your process for hardening**..................
......................................................................

# Design Abstracts

## How Much Ductility Is Necessary?

DESIGNERS and engineers seem to think that there is a direct relation between percentage elongation and the performance of metals under specific service conditions. In almost all cases the tendency is to demand as high an elongation as possible, frequently resulting in an impractical value. The thought seems to be that the larger the elongation the better the performance will be with regard to special stresses and applications. However, an engineer who will fight for a minimum value of 30 per cent for a given steel will accept fifteen for a critical heat-treated part such as spring steel. He will not worry over the fact that the part may be seriously stressed internally during the heat treating and may have so-called stress raisers left in the metal or sharp corners built into it during fabrication.

When malleable iron having 5 per cent elongation was produced it was acclaimed widely because of this value. Why? Because everyone was thinking of cast materials like gray iron which had practically zero elongation. Then when malleable was produced having 10 per cent stretch it was considered marvelous. This material was used under loading conditions where steel was required to have at least 20 per cent elongation. Aluminum and magnesium alloys were commonly used as castings with elongation of less than 5 per cent for important units such as parts of landing gears for airplanes. Now this material is universally accepted and thought to have excellent properties based on a value of less than 10 per cent.

Reason for this attitude can only be due to relative thinking. It would seem that designers and engineers might begin to question the practice of demanding as much elongation as the trade will allow and attempt to make specifications which more nearly meet the service conditions.—*From a paper by W. J. Conley, The Lincoln Electric Co., presented at the Twenty-Fifth annual meeting of A. W. S. in Cleveland.*

## Evaluating High-Strength Adhesives

BECAUSE there are many types of adhesives and many uses to which they may be put, no simple test yet devised seems to answer all requirements of a universal test. Two critical ways in which force can be exerted on a bonded surface are shear and direct tension. Naturally, in actual use, an adhesive may encounter either or any combination of the two. The combination is illustrated by the peelback or stripdown as when a flexible material is stripped from a rigid base.

Strength tests by no means tell the whole story, since the usage to which the adhesive will be put will determine the evaluation used. For instance, where two adhesives, each giving the same shear and stripdown values, are put under constant strain of low intensity, one may be satisfactory while the other will gradually fail. To determ this, a dead-load test is used in which a load below failure value is placed on the assembled bond. On other hand, some adhesives show good values when slow steady force is exerted, but fail under a sudden pull.

Conditions of use must also be evaluated since many hesives may show excellent properties when tested at mal temperatures, but may fail at elevated temperatures at very low temperatures. Permanence of the bond determined by aging the assembly in the Geer oven, oxygen bomb, or the air bomb. When the conditions such that the bond must withstand oils, solvents or wa it is well to make tests against these materials.

There are also specification tests which are used to sure maintenance of quality. These are usually rout tests, such as solids content to insure that an adhes is properly formulated, viscosity to insure that the terial will work properly, and strength tests to insure formity of results. These tests, as well as that for stabi in the container—usually by aging at elevated tempe tures—are usually run on each lot of adhesive made, order to make sure material of the correct grade is bei sent to the customer.—*From a paper by Fred Wehm Adhesives Division, Minnesota Mining & Mfg. Co., pr sented at the annual meeting of the A. S. M. E.*

## Combatting Moisture in Aircraft

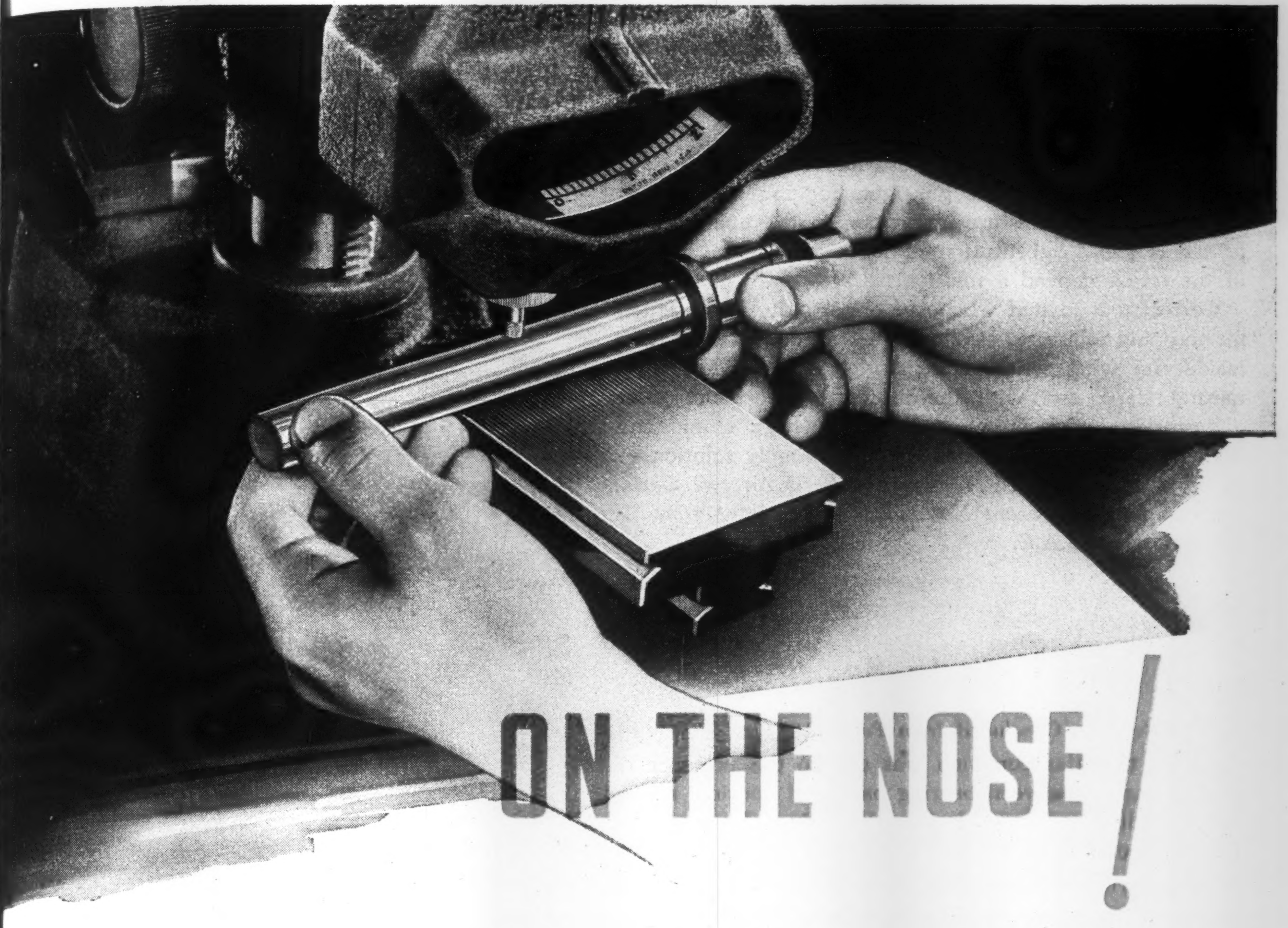
HUMIDITY has a damaging influence on an airpla and its equipment by setting up corrosion, produ ing mechanical interferences when ice is formed, or terfering with materials in general. Mechanical devic usually are sufficiently well lubricated so that corrosi of the working parts is minimized. However, corrosi occurs with loss of the lubricant or the exposure of protected parts, with resultant malfunction of the un Formation of ice may cause seizure or jamming of unit as well.

Serious corrosion, resulting in seizure of sealed b bearings on the carburetor push-pull rods, has been perienced. Difficulties of this nature can be lessened sealing all voids with the base lubricant—thereby p venting introduction of moisture by breathing—as well by periodic checks.

Some corrosion of the heart strand in cables has curred. Effective lubrication, periodic inspection or use of clad cable reduces this difficulty.

Cooling coils present a problem of plugging with due to the very nature of their installations. An expe enced pilot can correct this by closing the shutters wh he observes the temperature rising due to a plugged co rather than opening the shutters and freezing more.

Ventilation systems, dampers and gates are prone



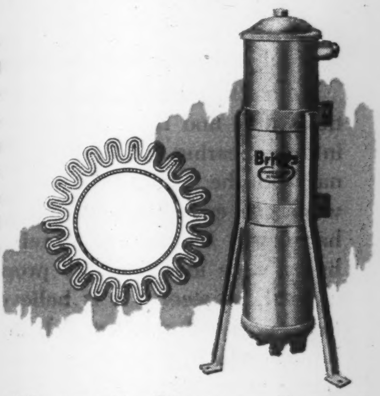
## Clean Coolant Helped...

Effective filtration of coolants or cutting oils may be the solution to your "percentage of rejects" problem. Reduce that percentage and you'll increase your production just that much!

Briggs Coolant Filters . . . with unique Z-fold refill cartridge . . . have shown remarkable results wherever they have been installed. On one machine, production was doubled . . . on another machine, increased production paid for the Briggs filter the first day it was installed. In addition—coolant life was doubled . . . wheel and tool life considerably extended.

Briggs Filters are available for unit machines or for central systems. Capacities range from 3 GPM to 100 GPM.

Look in your local classified telephone directory under "Filter" . . . locate the Briggs distributor . . . have him show you the difference between Briggs Filters and ordinary filters. Or—write direct to manufacturer for illustrated literature.



**Unique design and construction of Z-fold refill cartridge provides greatest possible filtering surface area and maximum flow rate. Cellulose is specially processed...from raw material to finished product . . . for filtration of liquids used as coolants.**

**BRIGGS CLARIFIER CO.** Briggs **General Offices, Washington 7, D. C.**

PIONEERS IN MODERN OIL FILTRATION

stick due to ice formation. Doors, sliding windshield panels, and emergency hatches are seriously affected by condensation and freezing. Careful design solves these problems.

Frequently moisture condenses on the inside of instrument glasses so that the instrument is invisible. In the case of one type of gunsight, fogging of the lens occurred due to moisture absorption in the fiber gaskets of the reticle. Substitution of lead gaskets and baking of the reticle proved a solution.

Corrosion is invited whenever opportunity is presented for trapping moisture. Condensate will collect in regions inside the wing where ribs, stringers, and fillets form natural traps, and with the presence of exhaust gases a corrosion action is set up. Basic design eliminating such traps or providing drainage are the only solutions—*From a paper by B. A. Rose, Lockheed Aircraft Corp., presented at the recent National West Coast Aeronautic meeting of the S.A.E.*

# Heavy-Duty Bearings

(*Concluded from Page* 116)

believed that this corrosion was due to the presence of foreign matter between bearing back and housing, resulting, initially, in a local decreased bearing clearance and local temperature rise.

**Bond Strength:** Poor bond between the bearing lining and the shell metal is a frequent cause of bearing failure and, in most cases, is due to defective manufacture. The heat flow through a poor bond area is usually impeded and may lead to failure because of uneven and excessive bearing-surface temperature. Fatigue cracks over poor bond area will loosen the lining quickly.

With a sound bond there are no inclusions or foreign matter in the joint. Such inclusions are usually due to imperfect cleaning of the shell surface before applying the bonding alloy. Poor bond in cases of trimetal or "sandwich" bearings may occur also between the intermediate bronze layer and steel of the shell. Nonmetallic inclusions between the bronze and steel will decrease the heat flow from the bronze to the steel, and local overheating may result. Areas of poor bond of a failed bearing are usually without any trace of bonding alloy.

**Foreign Particles:** Foreign, especially hard, particles may find their way into the lubricating-oil line and into the bearing and thus damage by scoring both the journal and the bearing surfaces. Such particles, however, would be embedded by soft bearing linings and become harmless. In one case foreign particles embedded in a lining were identified, by proper etching and by a small magnet, as steel. It is believed that they resulted from loosened weld splatters. It is now specified that special liquids which prevent adhesion of weld splatters to the frame must be used during welding of the frame. The use of such liquids is especially beneficial at corners and narrow passages inaccessible to the steel brush.

Thin white-metal bearing linings, .006 to .010-inch, have higher load-carrying capacity than thick linings, .015 to .125-inch, and are therefore more frequently used than thick linings. However, thin linings cannot fully embed
large foreign particles, and this will result in the scoring
the journal.

From the foregoing discussion of common bearing fa
ures it is evident that there is no optimum bearing materi
which will satisfy and comply with all requirements of
good bearing material. For this reason, selection of
bearing material is made by the designer with due co
sideration of all operating conditons, and therefore shou
result in the optimum combination of lining material, jou
nal hardness, lubricating oil, and others.

Maintenance and performance of precision-type bea

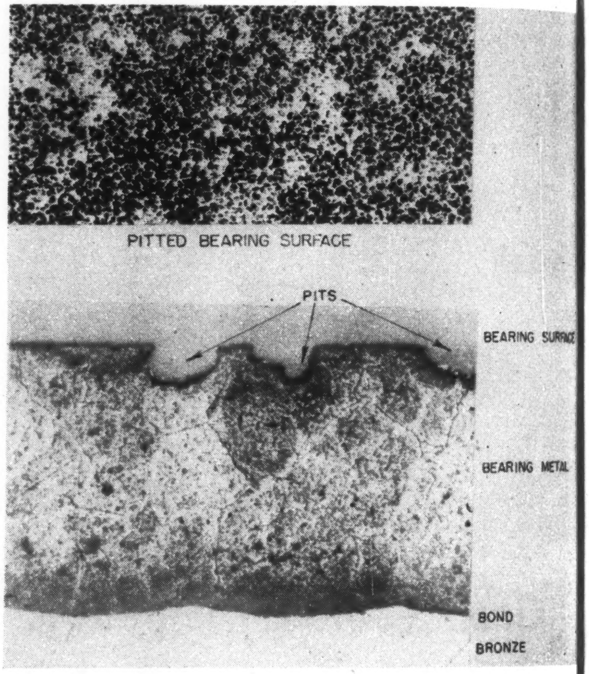
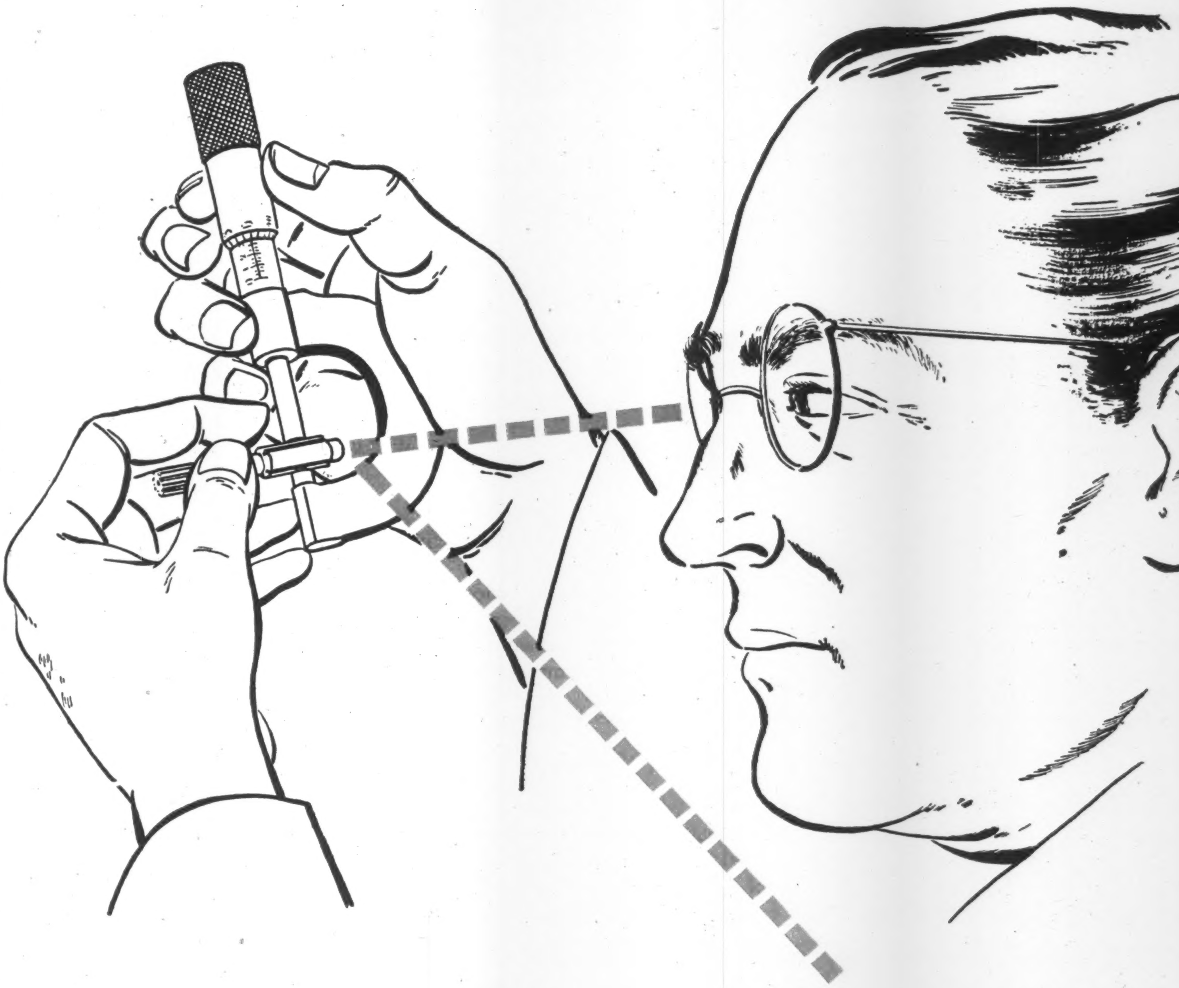


*Fig. 4—Pitted bearing surface due to corrosion. Uppe
view is approximately 4 times size, lower cross-sectio
is 54 times size. Surface pits are about .004-inch deep*

ings used in small engines, under 3-inch journal diamete
and in large engines, over 3-inch journal diameter, diffe
The manufacture, especially the grinding, of small crank
shafts is made with a high degree of accuracy which re
sults in uniformity of diameter and small tolerances. Bear
ings for such small journals, which for uniformity are ofte
diamond-bored, are interchangeable and do not requir
hand-fitting. The manufacture of a large crankshaft, o
the other hand, is not as accurate, resulting in large fina
tolerances. In addition, the surface finish of large jour
nals is not as smooth as that of a small journal. Bearing
for large journals might not fit the desired area, over 70 pe
cent contact, and are sometimes hand-fitted.

Use of approved low-corrosive lubricating oils is impor
tant, while adequate cooling of the lubricating oil wil
diminish the possibility of rapid oxidation. It is recom
mended that chemical analysis of the lubricating oil be
made at appropriate intervals—100 to 500 hours, depend
ing upon the service conditions—in order to check for dilu
tion and oxidation. It is also recommended that inspec
tion of sump-oil samples, drawn from the lowest point, be
made for metal particles at regular intervals, together with
the inspection of the sump-oil filter screen. Cleanliness
when handling bearings is highly important.



FINER PRECISION ASSURES GREATER RELIABILITY

- If the pumping of gasoline, water-alcohol or other fluids is important to you, then do not underestimate the value of high precision engineering in the pumps you select for your long range peacetime program.

The wide and varied experience of Romec in solving numerous pump problems for many years will enable us to serve your peacetime needs even better than before the war.

We salute the Navy and the Air Forces for having given us the opportunity to do an engineering and production job that exceeded expectations.

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ROMEC AIRCRAFT PRODUCTS

Romec

# BUSINESS AND SALES BRIEFS

AMONG other organization changes made by Western Gear Works of Seattle and Los Angeles is the advancement of Roy Crawshaw to the position of manager of engineering and sales. Mr. Crawshaw will make his headquarters in Seattle. Formerly he had been manager of plants in the Los Angeles district.

◆

Construction will begin as soon as WPB approval can be obtained on a new building for the General Electric Company's Research Laboratory. Located in suburban Niskayuna, about four and a half miles from the main plant and offices in Schenectady, it will afford some fifty per cent more space than present facilities provide.

◆

According to a recent announcement by Pittsburgh Steel Foundry Corp., Glassport, Pa., D. P. Morgan has joined the sales staff as assistant sales manager of the Philadelphia district. Prior to his appointment Mr. Morgan had been sales engineer for Hanna Stoker Co., Cincinnati.

◆

New sales representatives have been appointed by the Brush Development Co., Cleveland. These are: Burlingame Associates Ltd., N. Y., to cover the eastern New York and northern New Jersey area; Morris F. Taylor Co. of Silver Spring, Md., to handle the central Atlantic, southeast and sout states; Terwilliger Sales Co. of Kansas City, Mo., to c Iowa, Kansas, Nebraska and Missouri; and David M. Lee of Seattle to handle Washington, Oregon and Idaho.

◆

Election of P. R. Mork as executive vice president has b announced by Crane Co., Chicago. J. A. Dwyer has ceeded Mr. Mork as vice president in charge of sales branch houses. Succeeding Mr. Dwyer, in turn, is Lucien Moore as general manager of sales and branch houses.

◆

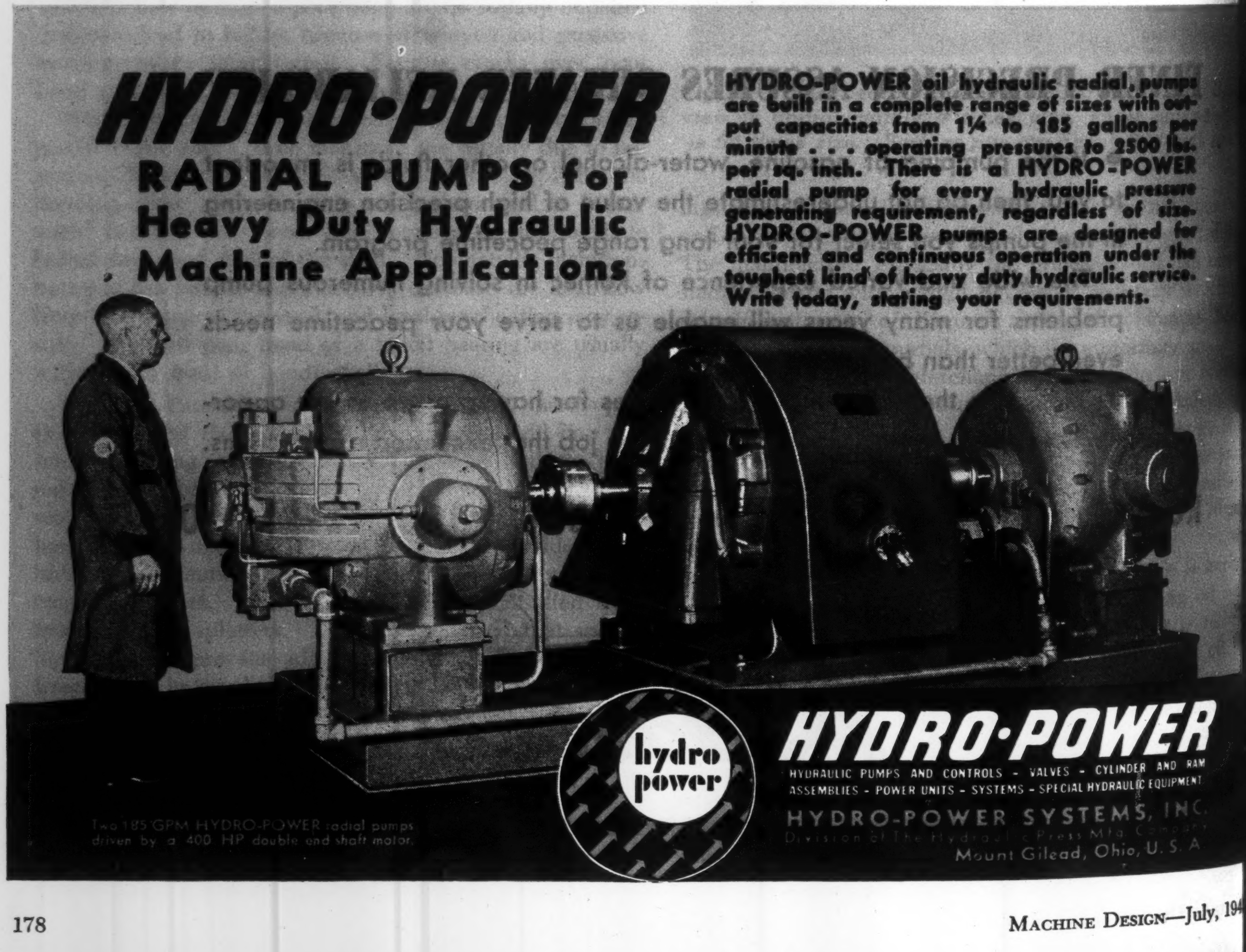
Arrangements have been completed by Stow Mfg. Binghamton, N. Y., to open a service branch at 1401 T street, San Francisco. Leon L. Allen will be in charge the new plant, while Don S. Macaulay, San Francisco reg representative, will continue to make his headquarters at Russ building.

◆

Included among recent appointments made by Robbins Myers Inc. are the following: F. W. Smith, motor sales m ager since 1929, has been elected vice president while W. Helmer, formerly assistant to the vice president in cha of sales, has been named general sales manager. Both make their headquarters at the home office in Springfield, O.

◆

Allis Chalmers Mfg. Co. has appointed J. C. Schaefer, f engineer of the Pittsburgh district office, as branch office m ager at Youngstown, O. In his new position Mr. Schae will succeed E. H. Legler, who has been transferred to electrical department at Milwaukee. Also announced is transfer of L. E. Stacey, field engineer of the Pittsburgh



HYDRO-POWER RADIAL PUMPS for Heavy Duty Hydraulic Machine Applications

HYDRO-POWER oil hydraulic radial, pumps are built in a complete range of sizes with output capacities from 1¼ to 185 gallons per minute . . . operating pressures to 2500 lbs. per sq. inch. There is a HYDRO-POWER radial pump for every hydraulic pressure generating requirement, regardless of size. HYDRO-POWER pumps are designed for efficient and continuous operation under the toughest kind of heavy duty hydraulic service. Write today, stating your requirements.

Two 185 GPM HYDRO-POWER radial pumps driven by a 400 HP double end shaft motor.

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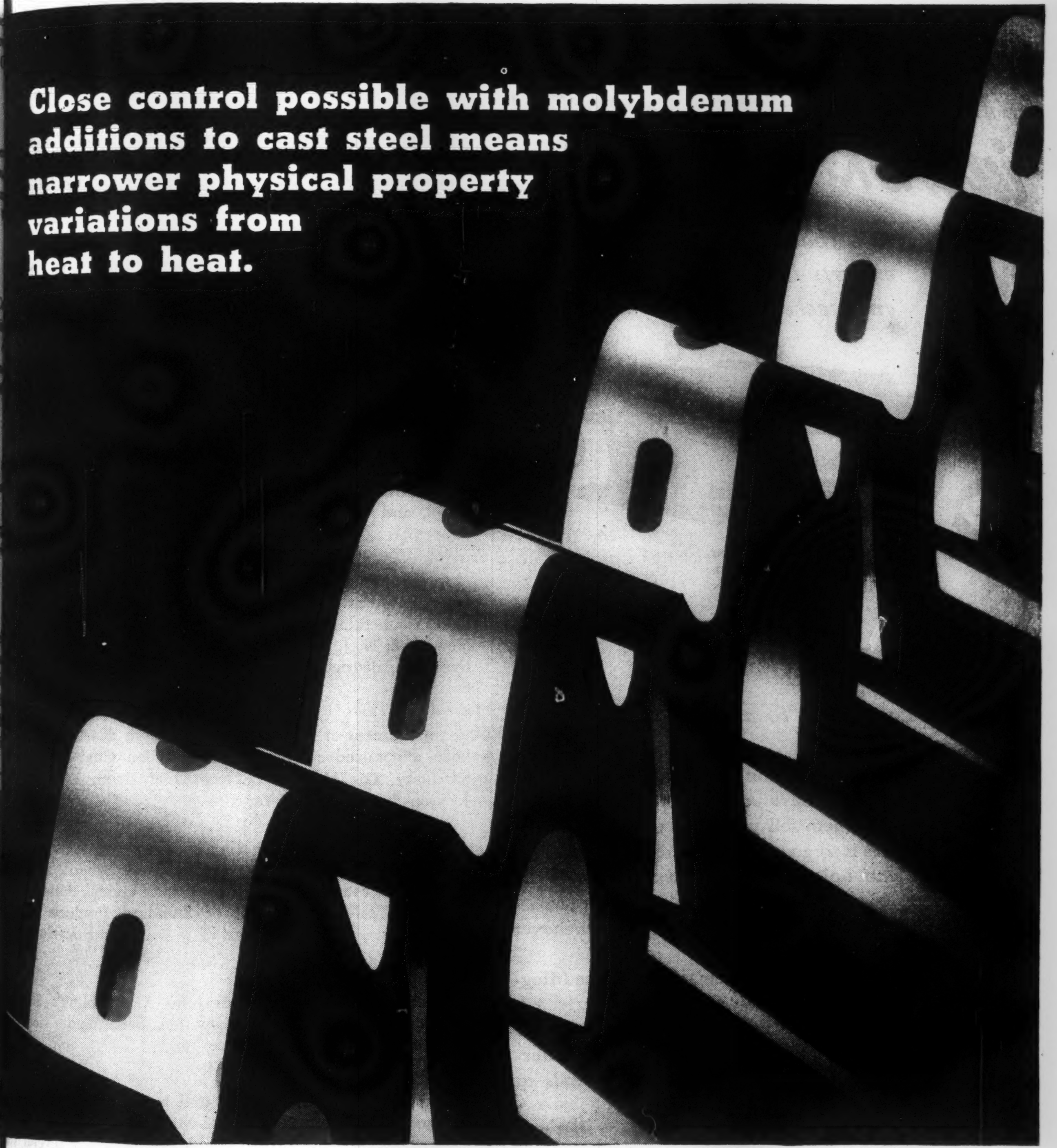
HYDRO-POWER

HYDRAULIC PUMPS AND CONTROLS - VALVES - CYLINDER AND RAM ASSEMBLIES - POWER UNITS - SYSTEMS - SPECIAL HYDRAULIC EQUIPMENT

HYDRO-POWER SYSTEMS, INC.
Division of The Hydraulic Press Mfg. Company
Mount Gilead, Ohio, U. S. A.


Close control possible with molybdenum additions to cast steel means narrower physical property variations from heat to heat.

LIMAX FURNISHES AUTHORITATIVE ENGINEERING
ATA ON MOLYBDENUM APPLICATIONS.

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MOLYBDIC OXIDE, BRIQUETTED OR CANNED •
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MOLY

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WHAT has Star GOT in a BRAKE MOTOR?

*. . . that convinces many experts it is the finest ever engineered?*

Specially designed Star Brake Motor for elevator duty on LST vessels.

*There are two ways to answer that one . . .*

We can give you a long list of makers of *cranes and hoists* who have used the Star Brake Motor . . . and let them tell you what there is about it that makes it so good . . . why they prefer STAR. Crane and hoist duty really tests out a brake motor! Name your field, if you'd prefer to check another – Star Motors are widely used whenever service is tough.

Or, we can give you the details about the Integral Disc Brake which Star designed and perfected. You'll see why this Brake is noted for *sureness* in operation . . . why it stays accurate and effective, *without adjustment* . . . why it leads to compact design.

There are plenty of superior features that you should know about before you pick a Brake Motor for your product or your production line. Write us for the facts. Star Electric Motor Co., 200 Bloomfield Ave., Bloomfield, N. J.

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STAR MOTORS

POWER PACKAGED AS YOU NEED IT

trict office, to Wheeling, W. Va., where he will serve resident field engineer. Both Mr. Schaefer and Mr. St will report to G. V. Woody, manager of the Pittsburgh trict office.

◆

Opening of offices in the Singer building, 149 Broadw New York 6, has been announced by Landis Tool Co Waynesboro, Pa. Walter P. Lotz, recently named e sales manager, has been placed in charge.

◆

Previously associated with Bakelite Corp., Dr. Nor A. Skow has assumed his new post as director of rese for Synthane Corp., Oaks, Pa., and will devote his eff to the research and development of Synthane plastics.

◆

Election of Colonel H. A. Toulmin Jr. as chairman of board and president has been announced by The Hydra Press Mfg. Co. of New York and Mt. Gilead, O. For president Howard F. MacMillin has been elected presid of a subsidiary of the company, The H-P-M Developm Corp., and will devote his attention to research work.

◆

Ampco Metal Inc., Milwaukee, has transferred L. P. Schru from the Accounts Service Department at Milwaukee to field engineering office at 1060 Broad street, Room Newark 2, N. J. Mr. Schrubey will serve as field represe tive under district manager W. T. Peterson.

◆

Appointment of A. C. Mohr as Chicago district manager been announced recently by The Union Chain & Mfg. Sandusky, O. He will be located at Room 1236, 53 W Jackson boulevard, Chicago.

◆

With the title of vice president and general sales m ager, D. R. Stamy has taken on the additional duties of m aging the sales department of Standard Products Co. will continue to be in charge of research and development

◆

Connected since 1910 with John A. Roebling's Sons Trenton, N. J., Ernest C. Low has been made vice pr dent in charge of sales.

◆

W. H. Ross has been appointed by Udylite Corp., Detr to serve as district manager of Michigan and northern O territory. Mr. Ross has covered this area for several ye as sales engineer for the company.

◆

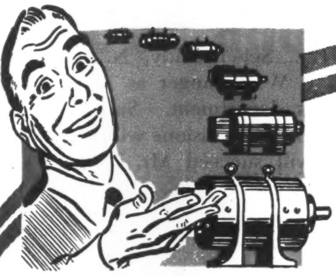
Associated with the company for twenty-six years, W. Ea Pashley has been elected president of C. F. Pease Co., Chica to succeed Thomas Lord who retired recently. Mr. Pashl has served as general sales manager, advertising manag treasurer and vice president.

◆

Made known recently by Jones & Laughlin Steel Corp. the following appointments: J. O. H. Anderson, formerly m ager of sales, tubular products, and H. B. Sparkman, previou district sales manager of the Philadelphia office, as assist general managers of sales. V. A. Jevon, assistant gene manager of sales, has been placed in charge of the Pittsbur district sales office. C. T. Hapgood, previously assistant ma

# What has Star got?

*Before you pick any motor to power your postwar product we'd like to submit 3 teasers . . .*



*(". . . and I know 'em all!")*

**Why did a famous maker of Materials Handling Equipment* not long ago say — "There isn't a Brake Motor built that can touch that Star, and I know 'em all"?**

*(". . . you like 'em tough")*

**Why did one of the world's biggest makers of Machine Tools* recently choose Star to work with on a tough application problem?**

*(". . . laying it in your lap")*

**Why did a noted Dairy Machinery company* pick Star to develop a unique small Generator for their new machine?**

(**Name on request*)

We needn't tell you that buyers like these three are motor-wise, and hard-boiled in their determination to obtain the most satisfactory solution to powering their products. . . .

And we needn't tell you that we have engineering brains second to none, or that we have excellent production facilities. Without them, we could not get to first base. . . .

What, then, is the "something extra on the ball" we *must* have to lead such concerns to choose to work with Star on their new products?

You'll find it's Star's approach to a motor problem – a willingness to give your job "top billing". There's no stop-watch on the brains applied to your problem – and there are not so many problems that we can't apply our best brains to all.

That's why you, too, will choose wisely by making Star your postwar "Motor Department". Our designing facilities can be placed at your command NOW, to solve any problems not covered by standard Star Motors, from 1/8 to 200 HP, some of which are available for early delivery. Star Electric Motor Co., 200 Bloomfield Ave., Bloomfield, N. J.

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Gasoline
Glue
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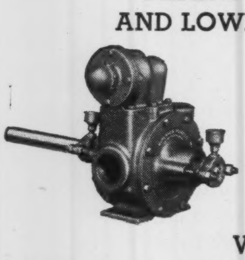
Rosin
Soaps
Solvents
Soups
Syrups
Tar
Yeast

BLACKMER ROTARIES
are SELF-ADJUSTING FOR WEAR

"Bucket Design" swinging vanes
automatically compensate for wear.

THIS MEANS SUSTAINED CAPACITY
DEPENDABLE OPERATION
AND LOWER PUMPING COSTS



WRITE FOR
Bulletin No. 306—Facts about Rotary Pumps
Bulletin No. 302—Pump Engineering Data

BLACKMER PUMP COMPANY
1970 Century Avenue Grand Rapids 9, Michigan

POWER PUMPS - HAND PUMPS
· EZY-KLEEN STRAINERS ·

ager of sales, tubular products, has been made manag sales, tubular products. Formerly district sales manag the Chicago office, E. W. Harwell has become district manager of the Philadelphia office. L. C. Berkey, previ district sales manager of the St. Louis office, has been na district sales manager of the Chicago office while C. C. W ling, formerly district sales manager of the Pittsburgh o has become district sales manager of the St. Louis office. has also been announced that the district office of Jone Laughlin Steel Corp. and the office of Jones & Laug Supply Co. have been moved to the Field building, 135 S LaSalle street, Chicago 3.

◆

According to a recent announcement by Sterling Ele Motors Inc., Arval Morris has been made national sales m ager and will be in charge of unifying and developing national sales program of the company.

◆

General Electric Co., Schenectady, N. Y., has appoi J. E. N. Hume and J. W. Belanger as assistant manager sales of the Apparatus department. Succeeding Mr. Hu as manager of the Industrial divisions will be Karl H. Run while R. S. Neblett will succeed Mr. Belanger as mana of the Federal and Marine divisions.

◆

Announced recently by the Pacific Division of Bendix A tion Corp., North Hollywood, Calif., is the opening of a sa engineering office in the Continental building in St. Lo Sales engineer Harold W. Bailey will handle the territ west of the Mississippi, while sales engineer William P. H rison will cover the area east of the Mississippi.

◆

Armon N. French has been named assistant sales mana of Wickwire Spencer Metallurgical Corp., subsidiary of Wi wire Spencer Steel Co., and will continue to be located the company's plant, 260 Sherman avenue, Newark 5, N. J

◆

Recently General Alloys Co., Boston, appointed Herb King of King & Anderson as field representative for the s of California, with offices at 625 Tilden Sales building, Market street, San Francisco.

◆

With headquarters at 1508 Rand building, Buffalo, Ne C. Taylor has been appointed Tenite sales representative Tennessee Eastman Corp. He has been connected with company for the past ten years, recently acting as Ten service engineer.

◆

Recently made known by Briggs Clarifier Co., Washingt D. C., is the appointment of E. Diggcs La Touche to m age the field service engineering staff of the Aviation Divisi

◆

General Controls Co. of Glendale, Calif., has named Le M. Horvath as divisional sales manager for the HI-G Divisi Recently Mr. Horvath had been associated with Adel Precis Products Corp. of Burbank, Calif.

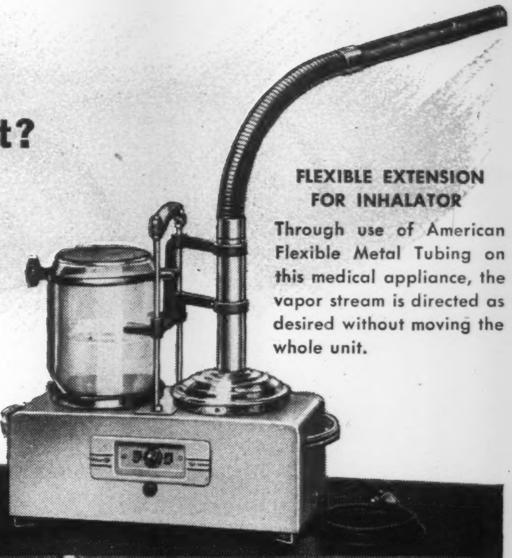
◆

According to a recent announcement by Fenwal Inc., A land, Mass., Carroll J. Robinson has replaced E. P. Grisw as sales and advertising manager.

## Looking for ways to improve your Product?

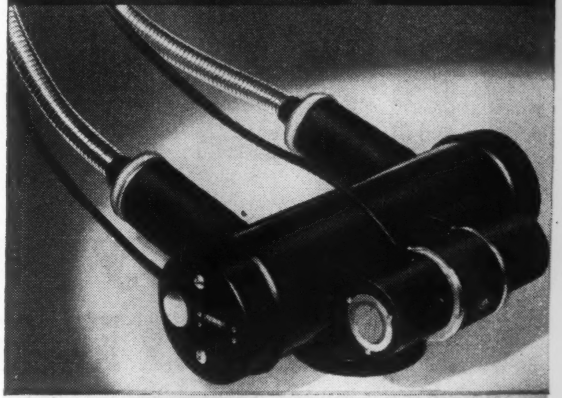
# Try *American*

# FLEXIBLE METAL HOSE and TUBING



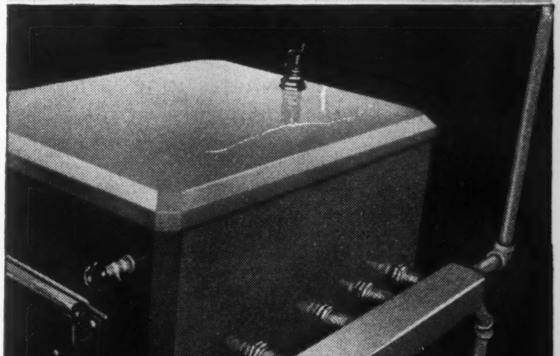
**FLEXIBLE EXTENSION FOR INHALATOR**

Through use of American Flexible Metal Tubing on this medical appliance, the vapor stream is directed as desired without moving the whole unit.



**PROTECTING X-RAY MACHINE CABLE**

Interlocked Stainless Flexible Metal Hose used to reinforce end of electric cable and prevent wiring failure.



**BOILER—"HEADER HEATER" CONNECTIONS**

Installation made quickly and easily through use of American Seamless Flexible Bronze Tubing connections.

WOULD YOU LIKE to make your product more convenient, safe or economical to install or operate?

Then look into the possibilities offered by American Flexible Metal Hose and Tubing.

Sturdy, leakproof connections of this material have improved literally thousands of types of products and equipment . . . from heavy machinery to small accessories. Illustrated are three examples . . . on each, flexible metal tubing is used to excellent advantage, yet for an entirely different purpose.

Find out where flexible connections in your product or equipment could better compensate for movement of parts, vibration and misalignment . . . or where gases, liquids or steam could be conveyed more easily or safely.

Flexible metal assemblies are usually easy to order and install. If *your* needs are unusual, our Technical Department can probably help.

45200

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